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COMPARISON OF HYDROGEN AND METHANE AS COOLANTS IN REGENERATIVELY COOLED PANELS

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16. Abstract

Comparisons are made of the minimum weights, ranges of applicability, and coolant requirements of methane-cooled, and hydrogen-cooled, structural panels determined from analytical studies of flat panels designed for heat fluxes up to 500 Btu/sec-ft^2 (568 kN/m^2) and pressure loads up to 250 psi (1720 kN/m^2).

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FOREWORD

This report was prepared by AiResearch Manufacturing Company, a division of The Garrett Corporation, Los Angeles, California, for the Langley Research Center of the National Aeronautics and Space Administration. This report presents the results of an analytical study performed under Task Order No. 5, "Comparison of Methane and Hydrogen as Coolants in Regeneratively Cooled Panels." The work is part of a comprehensive analytical and experimental study of regeneratively cooled panels performed under Contract NAS 1-5002. This program was under the cognizance of Dr. M. S. Anderson and Mr. J. L. Shideler of the Aerothermoelasticity Section and Mr. R. R. Howell and Mr. H. N. Kelly of the 8-Foot High Temperature Structures Tunnel Branch of the Structures Division, Langley Research Center.

COMPARISON OF HYDROGEN AND METHANE AS COOLANTS

IN REGENERATIVELY COOLED PANELS

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SUMMARY

An analytical study has been made of the weights and coolant requirements of methane- and hydrogen-cooled structural panels. The weights were based on design procedures for minimum weights developed under references 1 and 2. The present studies encompassed a range of heat fluxes from 10 to 500 Btu/sec-ft^2 (114 to 5680 kW/m^2), a range of applied pressures from 6.9 to 250 psi (48 to 1720 kN/m^2), and coolant outlet temperatures of 1400^0, 1600^0, and 1760^{0}R, (778^0, 889^0, and 978^{0}K). The results of the study indicate that the weight of methane required to accommodate a given heat flux will be 4.5 to 4.8 times that of hydrogen, but that the tankage volume for liquid methane will be 20 to 25 percent less than that for the liquid hydrogen. Pressure losses in the methane cooled panels were higher and thermal conductances were generally lower than those in the hydrogen cooled panels. Consequently, the methane cooled panels were generally slightly heavier than the hydrogen cooled panels and could not be designed to accommodate the higher heat fluxes at the higher coolant outlet temperatures.

INTRODUCTION

In recent years several studies have been made that indicate certain advantages in using liquid methane as the fuel for high-speed aircraft (see, for example, references 3 and 4). In general, these studies have dealt with aerodynamic and propulsion efficiency and have not evaluated the detail problems that may accompany its use. At hypersonic speeds regenerative cooling (fuel as coolant) is required over relatively large surface areas, particularly in the inlet and engine ducting. Possible effects of using methane on coolant requirements and structural weight must be assessed before its potentials and limitations as a fuel-coolant can be fully defined.

In the present report, an attempt has been made to compare directly the coolant flow requirements and the minimum weights of hydrogen-cooled and methane-cooled structural panels for a range of combinations of uniform heating and loading. The analytical procedures used for establishing minimum weight structures and minimum weight heat exchangers were those developed in references 1 and 2, respectively. Two cooled panel concepts were studied. One was an integrated concept wherein the heat exchanger was also the load-carrying panel structure. The other concept was a heat exchanger metallurgically bonded to a load-carrying panel.

In the study, the range of net heating was varied from 10 to 500 Btu/ft^2-sec (114 to 5680 kW/m^2) and the applied external pressure load was varied from about 7 to 250 psi (48 to 1720 kN/m^2). Minimum weights of panels designed for use with hydrogen and methane were compared for coolant outlet temperatures of 1400°, 1600°, and 1760°R (778°, 889° and 978°K) and for coolant inlet pressures up to 1000 psia (6890 kN/m^2) as required to provide a discharge pressure of 250 psi (1720 kN/m^2). The study was carried out primarily for a 2-ft by 2-ft (61-cm by 61-cm) panel that was shown in reference 1 to be a practical size for actively cooled panels.

SYMBOLS

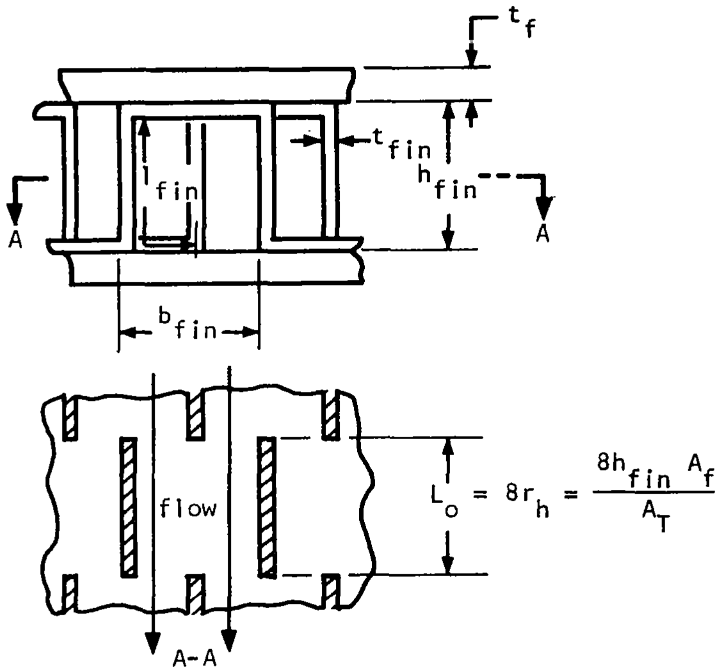
A	area exposed to heating, ft^2 (m^2)
A_f	minimum coolant flow area, ft^2 (m^2)
A_T	total heat transfer area on coolant side, ft^2 (m^2)
b	fin or web spacing, in. (cm)
b_F	flange width, in. (cm)
C_p	specific heat at constant pressure, Btu/lb-°R (J/g-°K)
f	friction factor
g_c	conversion factor, 32.2 ft/sec^2 (9.807 m/s^2)
H	enthalpy, Btu/lb (J/g)
h	height, in. (cm)
$\bar{h}$	heat transfer coefficient, Btu/sec-°R-ft^2 (kW/°K-m^2)
j	Colburn's modulus
k	thermal conductivity, Btu/hr-°R-ft (W/m-°K)
L_0	fin offset length, in. (cm)
l	panel length or coolant flow length, in. (cm)
l_{fm}	effective fin length, in. (cm)
N	number of fins/unit width, $N = \frac{1}{b_{fin}}$, in.$^{-1}$ (cm^{-1})
P	pressure, psi (kN/m^2)
P_R	Prandtl number
q	heat transfer rate, Btu/sec (kW)
r_h	hydraulic radius, in. (cm)

Re	Reynolds number
T	temperature, ^0R (^0K)
T_{DMW}	design maximum wall temperature, ^0R (^0K), $T_{DMW} = T_{CO} + \Delta T_{fin} + 2/3\ (\Delta T_f)$
t	thickness, in. (cm)
W	coolant flow rate, lb/sec (kg/s)
w	panel width or coolant flow width, in. (cm)
Δ	increment
η_0	overall heat transfer effectiveness
μ	coolant viscosity, lb/sec-ft (kg/s-m)
ρa	coolant density at average pressure and temperatures, lb/ft^3 (kg/m^3)
σ	ratio of density to unit density

Subscripts

C	coolant
c	core
F	flange
f	face sheet
fin	fin
I	inlet
O	outlet
W	web

Heat exchanger geometry nomenclature



Fin geometry is designated with a 4-part nomenclature:

20(7.9)R-0.10(0.25)-0.004(0.010)

- t_{fin}, in. (cm), fin thickness
- h_{fin}, in. (cm), fin height
- Designation of cross section and/or type of flow length interruption
  - R is for plain rectangular fins
  - RO is for rectangular offset fins
- Fin spacing $N = 1/b_{fin}$, fins/in. (fins/cm)

ANALYSIS AND METHODS

The objective of this study was to compare the overall structural unit weights and coolant flow requirements that result from designing hydrogen-cooled and methane-cooled minimum weight panels. Inasmuch as the methods for calculating the minimum weight cooled panels had been developed earlier (references 1 and 2), the major requirement to proceed with the design was to develop the necessary information and limitations for the use of methane as a coolant.

To initiate the study, the inlet conditions for the methane and its thermodynamic and transport properties had to be established. To determine a realistic methane inlet temperature, it was assumed that the methane picked up the same amount of heat between the storage outlet and panel inlet as the hydrogen did.

In the preceding study (reference 2), used in the present study for comparison purposes, the hydrogen inlet temperature was chosen to be $100^{\circ}R$ ($56\ ^{\circ}K$). The hydrogen was assumed to be stored as a saturated liquid at $40^{\circ}R$ ($22^{\circ}K$). Hence, the corresponding energy increase between storage and inlet was 268 Btu per lb (624 J/g) for an inlet pressure of 600 psi (4140 kN/m^2). Assuming a saturated liquid methane storage temperature of $214^{\circ}R$ ($119^{\circ}K$) and with the same energy increase, the methane inlet temperature was computed to be $287^{\circ}R$ ($159^{\circ}K$).

The transport properties of methane and hydrogen used in the present study are shown in figures 1 and 2, respectively. At the onset, the effects of pressure on transport properties were evaluated. It was found that the influence of pressure was noticeable only in the very low temperature range. Since the overall effect of pressure on the thermal conductance of the panels is small and the material properties of panels are not critical at low temperatures, it was decided that influence of variation of the transport properties with pressure on the heat exchanger design were not large enough to warrant the added complexity of attempting to account for it.

Two difficulties are encountered as a result of using methane as a coolant. First, at low coolant inlet pressures methane goes through a phase change from liquid to gas that results in significant variations in local heat transfer to the coolant. Second, at high temperatures methane cracks, or decomposes, freeing carbon which may deposit as a solid on the walls of the coolant passage resulting in increased wall temperature and in clogged passages.

A study was made of these two possible difficulties to determine the magnitude of their importance. For low heat flux cases the coolant inlet pressure required was not significantly greater than the outlet pressure of 300 psi (2070 kN/m^2). These pressures are below the critical pressure of methane and two-phase flow will occur over a length of the coolant passage. Figure 3 (a) presents a typical low pressure line on a pressure-enthalpy diagram and shows that the process follows essentially a straight line through the vapor dome. Figure 3(b) shows the corresponding heat transfer coefficient over the length of flow passage. In the region of the two-phase flow, there resulted a very large increase in the heat transfer coefficient. For an actual application the increase in the heat transfer coefficient would result in a minor increase in the coolant flow requirement. However, for the purposes of the present study a uniform net heat flux to the panel was assumed and the influence of phase change on coolant requirements was not considered.

The influence of the phase change on the structural design was found to be negligible. The net effect of the phase change, as shown by figure 3c, is to greatly reduce the local temperature level for the heated surface and to alter slightly the basically linear temperature distribution along the length of the cold wall (i.e. structural panel). The resulting reduction in the temperature differential between the heated surface and the cold wall (also shown in figure 3c) would tend to reduce the thermal stresses in the heated surface thereby increasing the fatigue life. However, due to the degradation of material properties with increasing temperature the critical design region occurs at the hot end of the panel and phase change has no net effect on the design of the heated surface. Similarly, the thermal stresses which are produced in the structural panel by the non-linearity of the temperature distribution occur in a non-critical area and do not influence the panel design.

To avoid serious difficulties associated with methane cracking, the methane outlet temperatures must be below that which will result in heat exchanger fouling by carbon deposition. A literature search was made to ascertain available data on cracking and deposition rates of methane flowing in passages. Figure 4 presents methane cracking reaction rates from two sources. The computational procedure outlined in reference 8 was used to estimate carbon deposit thickness for the present study with the assumptions that the methane was at outlet temperature for the last 20 percent of the passage length and that all of the carbon produced by cracking was deposited. These assumptions are all believed to be conservative. The period for carbon accumulation was chosen to be 100 hr. The calculations were made for the maximum heat flux level of 500 Btu/ft^2-sec (5680 kW/m^2) and for a 2-ft (61-cm) heat exchanger passage designed with an offset fin having 20 fins per inch width (7.9 fins per cm) and 0.003 in. (0.0076 cm) thickness in a passage 0.05 in. (0.13 cm) high. The results of the calculations for outlet temperature of 1800^{0}R (1000^{0}K) and 2000^{0}R (1110^{0}K) are presented in Table I, along with information concerning carbon film characteristics and methane flow.

The results indicate that for a period of up to 100 hr the effects of carbon deposition on heat exchanger performance are insignificant for outlet temperatures up to 1800^{0}R (1000^{0}K). At 2000^{0}R (1110^{0}K) the effects become noticeable. As a consequence, the design calculations for the methane-cooled structure were made for coolant outlet temperatures of 1400^0, 1600^0 and 1760^{0}R (778^0, 889^0, 978^{0}K).

Cooled Panel Concepts Studied

The two cooled panel concepts used in the present comparison of methane and hydrogen as coolants were developed in a general study of minimum weight regeneratively cooled panels, reference 1. In that study these concepts were shown to provide near optimum structural panels for specific ranges of combinations of heating and loading and are, therefore, considered to be representative of practical minimum weight regeneratively cooled panels. The procedures for achieving a minimum weight design for a specified combination of heating and loading is outlined for each of the concepts in references 1 and 2, as well as the process of, and the justification for, material selection. Therefore, the descriptions of the concepts that follow are brief.

Concept 1: a single-layered sandwich panel (figure 5). - This concept utilizes the capability of the sandwich panel to provide structural load-carrying capability, as well as flow passages for the coolant. Hence, the panel is designed to serve as a heat exchanger by carrying coolant internally while supporting a uniform externally applied pressure load. The heat exchanger has a straight-through, single-pass flow pattern and employs plain fins. The concept was selected for inclusion in this study, because it was shown in reference 1 to exhibit simplicity and light weight for low load-low heat flux application.

Inasmuch as the panel will be held flat by the back-up beams, temperature differences between the upper and lower panel face sheets will result in thermal stresses that will load the panel in the same way as the applied normal load. Hence, the sheets and fins must be sized to withstand internal coolant pressure stresses, normal pressure shear and bending stresses, as well as thermal stresses.

Concept 2: heat exchanger metallurgically bonded to prime panel (figure 6).- At higher loading conditions, it was shown in reference 1 to be advantageous to separate the structures related to load-carrying and thermal-protection functions. This separation of functions prevents the thermal stress and load stress from being additive. To meet this requirement with minimum weight, the heat exchanger was metallurgically bonded to the structural multiweb panel supported by beams. The heat exchanger has a simple, straight-through, single-pass flow pattern and employs a rectangular offset fin geometry.

Panel accessories. - A sketch showing the major accessories used in a detailed panel design is presented as figure 7. The sealing arrangement shown in figure 7(b) satisfies the requirement for allowing the panel to expand thermally while containing external pressure applied to one side of the panel. The manifold arrangement shown in figure 7(a) is the geometry using in making weight estimates. The attachment clips as illustrated in figure 7(c) are brazed to the inner side of the panels and bolted to the I-beams.

Materials. - The choice of materials is influenced by the heat exchanger performance in that structural working temperature is the primary factor governing the selection of materials. In the case of concept 1, where the composite structure operates at a relatively high temperature, the sandwich would be fabricated from Waspaloy. Inconel 718 was chosen for the back-up beams, attachment clips, and piping. Hastelloy X was used for the manifolds. For concept 2, Hastelloy X was chosen as the heat exchanger and manifold material. Inconel 718 was chosen as the prime load-carrying panel, back-up I beams, attachment clips, and piping material. As discussed more fully in reference 1, the material choices were based on conditions existing for a coolant outlet of

1600°R (889°K) and are not necessarily fully optimized at other coolant outlet temperatures.

Design restraints. - The same design limitations were imposed on both the methane- and hydrogen-cooled panel designs. The limitations included minimum gage restraints, as well as minimum coolant passage heights, minimum outlet pressure, maximum inlet pressure, maximum design temperature, and required life. The restraints are tabulated in Table 2.

Panel weight determination. - Tables 3 and 4 show typical weights as derived in reference 1 for panel concepts 1 and 2, respectively. The design conditions, materials, and material thicknesses are indicated. The coolant in both cases is hydrogen.

The geometric dimensions as provided by the minimum weight design procedures were used to determine the component weight per unit area. The component weights were then summed to get the total unit panel weight used in the comparisons that follow.

RESULTS AND DISCUSSION

Coolant Mass Flow Requirements

For the purposes of the present study uniform net heat fluxes were assumed over the surface of the panel. Under such conditions and with specified inlet and outlet temperatures the mass flow of coolant required is dependent only upon the heat capacity of the coolant.

Presented in figure 8 as functions of the net heat flux are the coolant requirements for methane and hydrogen for the three coolant outlet temperatures considered. The flow rates are based upon the transport properties and limitation presented in the section on analysis and methods. The weight of methane required to accommodate a given heat flux for a fixed outlet temperature is from 4.5 to 4.8 times that of hydrogen. However, as a result of differences in liquid density (liquid methane density = 26 lb/ft^3 (416 kg/m^3); liquid hydrogen density = 4.3 lb/ft^3 (69 kg/m^3)), the methane volume required to accommodate a given heat load is 20 to 25 percent less than the hydrogen volume. It should be noted, however, that if the coolant outlet temperature for the hydrogen cooled panel was allowed to rise to the limit set by the structural material (a condition which cannot be realized for the methane panel due to assumed coking limitations) or if the effects of phase change were considered (see analysis and methods section) the volumetric requirements of the methane cooled panels become less favorable.

Heat Exchanger Performance

The pressure drop and thermal conductance for the rectangular plain and offset fins used in the present study were calculated by use of the friction factor, f, and Colburn modulus, j, taken from reference 11 and shown as functions of Reynolds number in figure 9. The calculation procedure is described in reference 2. Results of typical calculations for one plain fin heat exchanger surface and one offset fin heat exchanger surface as presented in figures 10 and 11 respectively. Presented in the figures as functions of coolant flow rate are the pressure drops per unit length normalized to a density of 1 lb/ft^3 (16 kg/m^3) and the thermal conductances for methane and hydrogen.

Pressure drops. - It is apparent from figures 10 and 11 that at a given flow rate there is a little difference between the normalized pressure drops for the two coolants. However, when the differences in the densities of the two coolants and the flow rates required to accommodate a given heat flux are taken into account, the pressure drop through the methane heat exchanger is always considerably higher than that for the hydrogen heat exchanger. This is illustrated in figure 12 where the inlet pressure required for a 2-foot (0.61-m) long heat exchanger operating at a heat flux of 100 Btu/sec-ft^2 (1140 kW/m^2) and an outlet pressure of 300 psi (2070 kN/m^2) is presented as a function of fin height for three outlet temperatures. (The results presented in the figure include the pressure drop due to acceleration of the coolant in accordance with the procedure of reference 2. For hydrogen the increase in the pressure drop is 6 percent whereas, for methane the increase was approximately 4 percent.) At the minimum fin height (required for minimum weight) the heat exchanger inlet pressure for methane is from 1.25 to 1.45 times that for hydrogen. At higher heat fluxes or for longer panels the inlet pressure requirements for the methane cooled panels would be even greater relative to the hydrogen cooled panels since the pressure drops in the panels would be larger with respect to the fixed outlet pressure.

Typical manifold pressure drops (not a part of the pressure drop presented in figure 12) are shown in figure 13, where the combined inlet and outlet pressure drop for a concept I manifold is presented as a function of a parameter combining heat flux and the ratio of panel length to width. For the case shown, the methane pressure drop is approximately four times that for hydrogen.

Temperature differences. - Temperature differences through the depth of the heat exchanger are important from structural considerations because (1) they establish the thermal stress level in the hot surface, and (2) together with the coolant outlet temperature, they fix the maximum temperature at which the fins and hot surface will operate. For a given heat flux the temperature difference varies inversely with the thermal conductance of the heat exchanger. From figures 10 and 11 it can be seen that for a given coolant flow rate the temperature differences through a heat exchanger using methane as a coolant will be larger than those for hydrogen since the thermal conductance is always lower. The lower conductance is a direct result of the lower specific heat of methane.

For comparable heat fluxes and coolant outlet temperatures and with plain fins, such as used for the concept I panels, thermal conductance for methane varied from 90 to 150 percent of the values for hydrogen. Correspondingly the temperature differences and the design maximum wall temperatures for the methane heat exchanger were higher or lower than those for hydrogen depending upon heat flux as shown in figure 14. The variation shown in figure 14 follows directly from the variation of the Colburn modulus (j) during transition from laminar to turbulent flow (which occurs at Reynolds numbers from 3,000 to 10,000 as shown in figure 9) and the flow rates for the two coolants which places the methane, but not the hydrogen, flow in the transition range.

The effects of flow transition are not as pronounced for the offset fins which are used for the concept 2 panels. Thermal conductances for these fins vary in a more regular manner with the values for methane 70 to 90 percent of those for hydrogen at comparable heat fluxes and coolant outlet temperatures. Typical variations of temperature differences and design maximum wall temperature with heat flux for an offset fin are presented in figure 15. From a comparison of figures 15 and 14 (or indirectly from figure 9) it can been seen that that the temperature differences and design maximum wall temperatures are always lower for the offset fin configuration.

The effects of fin height on design maximum wall temperature and inlet pressure are shown in figure 12. The effect of fin height on temperature difference is indicated in figure 16. Increasing fin height, which increases weight but reduces pressure drop, also increases the temperature differences through the exchanger. This is as would be expected, since for a fixed coolant flow rate thermal conductance goes down with increased coolant passage size. The temperature difference can be reduced for a fixed fin height by increasing the number of fins per unit of width, by using thicker fins, or by using a material with a higher conductivity. However, none of these changes is accomplished without an increase in pressure drop and weight.

Temperature difference through the exchanger and the hot surface temperature level control the thermal stress level and low cycle fatigue life of the exchanger as well as influence its weight. It is clear that methane heat exchangers will be inferior to hydrogen heat exchangers except for the case of plain fins at conditions where the thermal conductance of methane is improved by transition and the hydrogen is not.

Total Panels Weight Comparisons

For each heating, loading, and coolant outlet temperature combination chosen, the minimum weight heat exchanger-structural panel was established for each of the two concepts considered using methane as the coolant. The results of these calculations are tabulated in Tables 5 and 6 for concept 1 and concept 2, respectively. Unit area weights for the two concepts are presented in figure 17 and compared with corresponding results for hydrogen cooled panels in figures 18 and 19.

Methane cooled panels - concepts 1 and 2. - The results for the methane cooled panels presented in figure 17 exhibit trends which are similar to those for hydrogen cooled panels reported in reference 1. The data indicate that the unit area weights are strong functions of the pressure loading and, with the exception of the concept 1 panels at heat fluxes between 10 and 50 Btu/sec-ft^2 (114 and 568 kW/m^2), weak functions of the heat flux. Concept 1 provides the lighter weight designs in the low pressure range, and concept 2 provides the lighter configurations at the higher loading conditions.

In contrast to the results for hydrogen cooled panels which indicate that for minimum weight designs the use of concept 1 must be restricted to low heat fluxes, the data presented in figure 17 indicate that for methane cooled panels the regions where concept 1 provides the lighter weight design are not limited to the very low heat flux range. The lighter weight of the concept 1 panels at higher heat fluxes, is linked to the occurrence of turbulent flow in the coolant passages. (As discussed in the section on heat exchanger performance, turbulent flow, which occurs at the flow rates required for the methane-cooled panels, increases the thermal conductance of the plain fin configurations. The higher thermal conductance, in turn, reduces the operating temperature of the structural material and permits the attainment of lighter weight designs.) Since the occurrence of turbulent flow is dependent upon the coolant flow rate, the effects on configuration weight are encountered at different heat flux levels for different outlet temperatures. The effect of turbulent flow on configuration weight in the data of figure 17 is most pronounced at the lowest coolant outlet temperature. (Turbulent flow is responsible for the increase in the range of pressures for which Concept 1 provides the lighter weight design shown in figure 17a for heat fluxes above 50 Btu/ft^2-sec (568 kW/m^2). It would be expected that for the higher coolant outlet temperatures (figures 17b and c) similar beneficial effects of turbulence on the concept 1 configuration weights would be experienced at heat fluxes slightly higher than those investigated.

The primary effect of increased coolant outlet temperature, other than the expected general increase in configuration weight, is that as the outlet temperature is increased the maximum heat flux for which design solutions can be obtained diminishes as shown by the results for concept 2 in figure 17. The figure does not clearly define the entire design solution boundary, however, since the maximum heat flux for which a design can be achieved was not established for each normal pressure. At the 1600^{0}R (899^{0}K) coolant temperature, it is indicated that no design was found for heat fluxes of 500 Btu/ft^2-sec (5680 kW/m^2) and at 1760^{0}R (978^{0}K) outlet temperature no design was found for 250 or 500 Btu/ft^2-sec (2840 or 5680 kW/m^2). It is thus apparent that for many conditions the lack of design solutions places more severe restrictions on the maximum usable methane outlet temperature than the somewhat arbitrary temperature limitation imposed in this study to avoid thermal decomposition of the methane.

Hydrogen and methane cooled panels. - Within the range of heating and loading where a particular concept was best there was no case where the differences between the weights of the methane and hydrogen cooled panels were extremely large. The maximum weight difference within this range was encountered with the concept 2 design (see figure 19) for a heat flux of 500 Btu/ft^2-sec (5680 kW/m^2), an external load of 50 psi (345 kN/m^2), and at a coolant outlet temperature of 1400^{0}R (778^{0}K). At this condition, the methane-cooled panel was about 8 percent heavier than the corresponding hydrogen panel. Larger differences are shown for the concept 1 design in figure 18, but these differences occur for conditions where concept 1 is heavier than concept 2.

For some conditions of heating and loading as shown in figure 18 the concept 1 methane cooled panels were slightly lighter than the corresponding hydrogen cooled panels. This is a consequence of the effects of turbulent flow, previously mentioned, which occurs at the flow rates required for methane but is not encountered at the flow rates required for hydrogen. Because transition from laminar to turbulent flow occurs within the flow range of interest for the methane cooled panels the weights and consequently the difference in weights between the methane and hydrogen cooled panels depend strongly upon heat flux and coolant outlet temperature.

For concept 2 the hydrogen cooled panels were found to be consistently lighter than the methane cooled panels (see figure 19) and the differences in weight were relatively insensitive to the variations in heat flux and coolant outlet temperature. The lower weight follows from the lower pressure losses and the higher thermal conductances (hence lower fin and face plate temperatures) obtained with hydrogen as a coolant. The lack of sensitivity of the differences in weight is attributed to the relative insensitivity of the thermal conductance and pressure loss of the offset fin to changes from laminar to turbulent flow, and to the fact that weight differences were in the heat exchanger and manifolds which represent only a small portion of the total weight for the concept 2 designs but a large portion of the total weight for the concept 1 designs.

CONCLUSIONS

An analytical study has been made of the differences in the regeneratively cooled panel weight and coolant flow requirements that are associated with the use of methane and hydrogen as coolants. The structural weight comparisons are based on a minimum panel and heat exchanger weight analysis procedure developed in references 1 and 2. The study was carried out over a range of heat fluxes from 10 to 500 Btu/ft^2-sec (114 to 5680 kW/m^2) and a range of applied pressure loading from 6.9 to 250 psi (48 to 1720 kN/m^2) with coolant outlet temperatures of 1400^0, 1600^0, and 1760^{0}R (778, 889^0 and 978^{0}K.) The more important conclusions are as follows:

(1) As a result of the differences in the thermodynamic and transport properties of the two coolants, the weight of methane required to accommodate a given heat flux is from 4.5 to 4.8 times that of hydrogen. Because of differences in density, the volume of liquid methane tankage required to accommodate a given heat load is 75 to 80 percent of that for hydrogen.

(2) The greater flow requirements more than offset the higher density of methane relative to that of hydrogen and result in greater heat exchanger and manifold pressure drops. The greater design containment pressure required to afford a given outlet pressure generally results in a heavier heat exchanger than that required for hydrogen. As a consequence, except for a very limited range of conditions, methane-cooled panels were found to be heavier than hydrogen-cooled panels. Differences in weights of the methane-and hydrogen-cooled panels for the minimum weight designs did not exceed 8 percent.

(3) The lower thermal conductance of methane relative to hydrogen results in greater temperature differences through the heat exchanger and a higher working temperature for the hot face of the heat exchanger. As a consequence, methane-cooled heat exchanger design solutions could not be obtained at the higher outlet temperatures and heat flux levels.

(4) Limited analysis indicated that, within the range of the present investigation, the effects of liquid-to-gas phase change and thermal decomposition of the coolant on the structural design of the methane-cooled panels were negligibly small.

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TABLE I

DEPOSITION OF CARBON FOR A 100-HR PERIOD

Panel Conditions		
Length = 2 ft (61 cm)		
Length of uniform carbon deposit = 0.4 ft (12 cm)		
Offset fin geometry = 20(7.9)R0-0.050(0.127)-0.003(0.0076)		
Duration = 100 hr		
Heat flux = 500 Btu/sec-ft^2 (5680 kW/m^2)		
Carbon Film Properties		
Thermal conductivity = 2.42 Btu/hr-ft-^{0}R (3.88 W/m-^{0}K)		
Density = 0.0749 lb/in.^{3} (2070 kg/m^3)		
Methane Conditions		
Inlet temperature = 287^{0}R (159^{0}K)		
Outlet pressure = 300 psi (2070 kN/m^2)		
Outlet temperature, ^{0}R (^{0}K)	1800 (1000)	2000 (1110)
Flow rate, lb/sec-ft (kg/s-m)	0.724 (1.08)	0.632 (0.942)
Outlet density, lb/ft^3 (kg/m^3)	0.249 (3.98)	0.224 (3.58)
Mol fraction cracked/sec	3.5×10^{-5}	4.7×10^{-4}
Results of Analysis		
Carbon deposition rate, lb/sec-ft (kg/s-m)	9×10^{-9} (13.4×10^{-9})	109×10^{-9} (16.2×10^{-9})
Deposited carbon film thickness, in. (cm) (accumulation for 100 hr)	0.00021(0.000534)	0.00254(0.00646)
Temperature differential through carbon film, ^{0}R (^{0}K)	13(7.2)	157(87)
ΔP with carbon/ΔP clean	1.014	1.19

TABLE 2

HEAT EXCHANGER DESIGN LIMITATIONS

Minimum fin thickness	0.003 in. (0.0076 cm)
Minimum fin height	0.025 in. (0.0635 cm)
Maximum fins per inch of width (fins per cm)	40 (15.8)
Minimum fins per inch of width (fins per cm)	20 (7.9)
Maximum inlet pressure	1000 psi (6900 kN/m^2)
Heat exchanger outlet pressure	300 psi (2070 kN/m^2)
Manifold outlet pressure	250 psi (1720 kN/m^2)
Fin thermal conductivity	10 Btu/hr-ft-^{0}R (17.3 W/m^0-K)
Hot wall thermal conductivity	14.5 Btu/hr-ft-^{0}R (25.1 W/m^0-K)
Maximum design temperature	1540^{0}F (1110^{0}K)
Heat exchanger life	100 hr
Hot wall thickness	0.010 in. (0.025 cm)
Hot gas recovery temperature	Infinite
Coolant flow length (except as noted on tables 5 and 6)	2 ft (61 cm)

STRUCTURAL DESIGN LIMITATIONS

Minimum web thickness	0.003 in. (0.0076 cm)
Minimum face sheet thickness	0.010 in. (0.0254 cm)

TABLE 3

GEOMETRIC PROPORTIONS AND MATERIALS OF STRUCTURAL ELEMENTS AND WEIGHT SUMMARY FOR CONCEPT I FOR THE FOLLOWING CONDITIONS

ℓ = 2 ft (0.61 m)

w = 2 ft (0.61 m)

Coolant = hydrogen

Coolant inlet pressure = 300 psi (2070 kN/m^2)

Coolant outlet temperature = 1600°R (889°K)

Normal pressure = 6.95 psi (48 kN/m^2)

Uniform heat flux = 10 Btu/sec-ft^2 (114 kW/m^2)

Fin conductivity = 10 Btu/ft-hr-°R (17 W/m-°K)

Waspaloy panel	*Inconel 718 beams*	*Inconel 718 attachment clips*
h_{fin} = 0.075 in. (0.191 cm)	h = 1.46 in. (3.70 cm)	Developed length = 2.61 in. (6.61 cm)
b_{fin} = 0.050 in. (0.127 cm)	b_F = 0.605 in. (1.54 cm)	t = 0.010 in. (0.025 cm)
t_f = 0.010 in. (0.025 cm)	t_F = 0.035 in. (0.088 cm)	Beam spacing = 7.77 in. (0.198 m)
t_{fin} = 0.003 in. (0.0076 cm)	t_w = 0.027 in. (0.068 cm)	
Wt = 1.27 lb/ft^2 (6.20 kg/m^2)	Wt = 0.46 lb/ft^2 (2.24 kg/m^2)	Wt = 0.18 lb/ft^2 (0.88 kg/m^2)

Manifolding			Hastelloy X seal
Hastelloy X inlet	Hastelloy X outlet	Inconel 718 piping	
ℓ = 3.25 in. (8.15 cm)	ℓ = 3.25 in. (8.15 cm)	t = 0.030 in. (0.076 cm)	Average thickness = 0.0130 in. (0.033 cm)
h_{fin} = 0.025 in. (0.063 cm)	h_{fin} = 0.025 in. (0.063 cm)	Diam = 1.75 in. (4.44 cm)	Width = 1.30 in. (3.3 cm)
b_{fin} = 0.100 in. (0.25 cm)	b_{fin} = 0.100 in. (0.25 cm)		
t_{fin} = 0.003 in. (0.0076 cm)	t_{fin} = 0.003 in. (0.0076 cm)		
t_f = 0.010 in. (0.025 cm)	t_f = 0.010 in. (0.025 cm)		
Wt = 0.14 lb/ft^2 (0.68 kg/m^2)	Wt = 0.14 lb/ft^2 (0.68 kg/m^2)	Wt = 0.30 lb/ft^2 (1.45 kg/m^2)	
Total manifold wt = 0.58 lb/ft^2 (2.83 kg/m^2)			Wt = 0.06 lb/ft^2 (0.29 kg/m^2)

Total weight = 2.55 lb/ft^2 (12.5 kg/m^2)

Coolant flow rate = 0.00187 lb/sec-ft^2 (0.00915 kg/s-m^2)

TABLE 4

GEOMETRIC PROPORTIONS AND MATERIALS OF STRUCTURAL ELEMENTS AND WEIGHT SUMMARY FOR CONCEPT 2 FOR THE FOLLOWING CONDITIONS

ℓ = 2 ft (0.61 m)

w = 2 ft (0.61 m)

Coolant = hydrogen

Coolant inlet pressure = 630 psi (4340 kN/m^2)

Coolant outlet temperature = 1600°R (889°K)

Normal pressure = 100 psi (689 kN/m^2)

Uniform heat flux = 250 Btu/sec-ft^2 (2840 kW/m^2)

Fin conductivity = 10 Btu/ft-hr-°R (17 W/m-°K)

Hastelloy X heat exchanger	Inconel 718 prime panel	Inconel 718 beams
h_{fin} = 0.027 in. (0.069 cm)	h = 0.293 in. (0.745 cm)	h = 3.34 in. (8.48 cm)
b_{fin} = 0.050 in. (0.127 cm)	b_f = 0.258 in. (0.656 cm)	b_F = 1.22 in. (3.09 cm)
t_f = 0.010 in. (0.025 cm)	t_f = 0.010 in. (0.0254 cm)	t_F = 0.072 in. (0.183 cm)
t_{fin} = 0.003 in. (0.0076 cm)	t_c = 0.0052 in. (0.0132 cm)	t_w = 0.053 in. (0.135 cm)
Wt = 0.72 lb/ft^2 (3.51 kg/m^2)	Wt = 1.11 lb/ft^2 (5.42 kg/m^2)	Wt = 3.19 lb/ft^2 (15.6 kg/m^2)

Manifolding			Seals	Attachment clips
Hastelloy X inlet	Hastelloy X outlet	Inconel 718 piping		
h_{fin} = 0.050 in. (0.127 cm)	h_{fin} = 0.142 in. (0.361 cm)	Diameter = 1.75 in. (4.4 cm)	Width = 1.30 in. (3.3 cm)	Developed width = 3.26 in. (8.18 cm)
b_{fin} = 0.100 in. (0.254 cm)	b_{fin} = 0.100 in. (0.254 cm)	Thickness = 0.030 in. (0.076 cm)	Average thickness = 0.049 in. (0.124 cm)	Thickness = 0.010 in. (0.025 cm)
t_f = 0.012 in. (0.031 cm)	t_f = 0.0135 in. (0.0343 cm)			Beam spacing = 4.70 in. (0.120 m)
ℓ = 3.25 in. (8.25 cm)	ℓ = 3.25 in. (8.25 cm)			
Wt = 0.19 lb/ft^2 (0.93 kg/m^2)	Wt = 0.26 lb/ft^2 (1.27 kg/m^2)	Wt = 0.30 lb/ft^2 (1.46 kg/m^2)		
Total manifolding wt = 0.75 lb/ft^2 (3.66 kg/m^2)			Wt = 0.23 lb/ft^2 (1.12 kg/m^2	Wt = 0.37 lb/ft^2 (1.80 kg/m^2)

Total weight = 6.37 lb/ft^2 (31.1 kg/m^2)

Coolant flow rate = 0.0468 lb/ft^2-sec(0.228 kg/s-m^2)

TABLE 5

MINIMUM WEIGHT CONCEPT I PANEL WEIGHTS FOR SELECTED HEATING, LOADING, AND COOLANT OUTLET TEMPERATURE CONDITIONS

(a) U.S. CUSTOMARY UNITS

ℓ, ft	T_{CO}, °R	q/A, Btu/sec-ft²	p, psi	Fin Geometry: N, fins/in.	Fin Geometry: h_{fin}, in.	Fin Geometry: t_{fin}, in.	P_{CI}, psi	T_{DMW}, °F	$T_{DMW}-T_{CO}$, °F	Panel wt, lb/ft²	Beam wt, lb/ft²	Clip wt, lb/ft²	Manifold wt, lb/ft²	Seal wt, lb/ft²	Total wt, lb/ft²	Methane rate, lb/sec-ft²
2	1400	10	6.95	20R-	0.050-	0.003	300.2	1141	201	1.20	0.50	0.23	0.59	0.06	2.58	0.01045
2	1600	10	6.95	20R-	.050-	.003	300.2	1328	188	1.20	.51	.23	.59	.06	2.59	.00872
2	1760	10	6.95	20R-	.050-	.003	300.2	1475	175	1.20	.56	.30	.59	.07	2.72	.00766
2	1400	50	6.95	20R-	.050-	.003	302.9	1270	330	1.20	.52	.25	.70	.06	2.73	.0509
2	1600	50	6.95	20R-	.025-	.003	316.6	1338	198	1.14	.56	.32	.70	.06	2.78	.0436
2	1760	50	6.95	20R-	.025-	.003	313.6	1517	217	1.14	.63	.43	.70	.07	2.97	.0383
2	1400	100	6.95	20R-	.050-	.003	310.8	1355	415	1.20	.53	.25	.71	.06	2.75	.1045
2	1600	100	6.95	20R-	.025-	.003	355.9	1392	252	1.14	.58	.34	.71	.06	2.83	.0872
2	1760	100	6.95	30R-	.025-	.003	362.5	1519	219	1.17	.64	.43	.71	.07	3.02	.0766
2	1400	10	50	20R-	.150-	.003	300.0	1215	275	1.46	2.20	.42	.59	.16	4.83	.01045
2	1600	10	50	30R-	.150-	.003	300.1	1295	155	1.65	2.12	.37	.59	.16	4.89	.00872
2	1760	10	50	30R-	.150-	.003	300.1	1443	143	1.65	2.31	.48	.59	.17	5.20	.00766
2	1400	50	50	20R-	.100-	.003	300.5	1560	620	1.33	2.50	.58	.70	.16	5.27	.0509
2	1600	50	50	40R-	.050-	.003	303.7	1423	283	1.33	2.69	.73	.70	.16	5.61	.0436
2	1760	50	50	40R-	.050-	.003	303.0	1604	304	1.33	3.09	1.08	.70	.17	6.37	.0383
2	1400	100	50	30R-	.100-	.003	303.1	1464	524	1.46	2.47	.52	.71	.16	5.32	.1045
2	1600	100	50	40R-	.050-	.003	316.6	1434	294	1.33	2.73	.77	.71	.16	5.70	.0872
2	1760	100	50	40R-	.050-	.003	313.3	1619	319	1.33	3.13	1.12	.71	.17	6.46	.0766
2	1400	10	100	30R-	.150-	.003	300.1	1110	170	1.65	3.79	.56	.59	.23	6.82	.01045
2	1600	10	100	30R-	.150-	.003	300.1	1295	155	1.65	3.77	.55	.59	.23	6.79	.00872
2	1760	10	100	40R-	.150-	.003	300.1	1399	99	1.84	3.96	.64	.59	.24	7.27	.00766
2	1400	50	100	40R-	.150-	.003	300.4	1414	474	1.84	4.03	.67	.70	.23	7.47	.0509
2	1600	50	100	40R-	.150-	.003	300.5	1586	446	1.84	4.48	.89	.70	.23	8.14	.0436
2	1760	50	100	40R-	.050-	.003	303.0	1604	304	1.33	5.51	1.59	.70	.24	9.37	.0383
2	1400	100	100	40R-	.100-	.003	304.1	1396	456	1.59	4.28	.78	.71	.23	7.59	.1045
2	1600	100	100	40R-	.050-	.003	316.6	1434	294	1.33	4.88	1.13	.71	.23	8.28	.0872
2	1760	100	100	40R-	.050-	.003	313.3	1619	319	1.33	5.58	1.66	.71	.24	9.52	.0766
5	1400	10	6.95	20R-	.050-	.003	300.7	1073	133	1.20	.49	.22	.42	.04	2.37	.01045
5	1600	10	6.95	20R-	.050-	.003	300.5	1284	144	1.20	.50	.22	.42	.04	2.38	.00872
5	1760	10	6.95	20R-	.050-	.003	300.5	1441	141	1.20	.54	.28	.42	.04	2.48	.00766
5	1400	50	6.95	20R-	.050-	.003	315.3	1124	184	1.20	.50	.22	.61	.04	2.57	.0509
5	1600	50	6.95	20R-	.050-	.003	313.3	1327	187	1.20	.50	.23	.61	.04	2.58	.0436
5	1760	50	6.95	20R-	.050-	.003	311.7	1491	191	1.20	.56	.30	.61	.04	2.71	.0383
5	1400	100	6.95	20R-	.050-	.003	353.1	1173	233	1.20	.51	.23	.62	.04	2.60	.1045
5	1600	100	6.95	20R-	.050-	.003	344.3	1381	241	1.20	.52	.24	.62	.04	2.62	.0872
5	1760	100	6.95	20R-	.050-	.003	338.9	1547	247	1.20	.58	.33	.62	.04	2.77	.0766

NOTE: Panel width, w = 2 ft

TABLE 5 (Concluded)

(b) SI UNITS

ℓ, ft	T_{CO}, °K	q/A, kW/m²	p, kN/m²	Fin Geometry: N, Fins/cm – h_{fin}, cm – t_{fin}, cm	P_{CI}, kN/m²	T_{DMW}, °K	$T_{DMW}-T_{CO}$ °K	Panel wt, kg/m²	Beam wt, kg/m²	Clip wt, kg/m²	Manifold wt, kg/m²	Seal wt, kg/m²	Total wt, kg/m²	Methane rate, kg/s-m²
0.61	778	114	48	7.9R-0.127-0.0076	2070	900	112	5.85	2.44	1.12	2.88	0.29	12.6	0.0510
.61	889	114	48	7.9R- .127- .0076	2070	993	104	5.85	2.49	1.12	2.88	.29	12.6	.0425
.61	978	114	48	7.9R- .127- .0076	2070	1075	97	5.85	2.73	1.46	2.88	.34	13.3	.0374
.61	778	568	48	7.9R- .127- .0076	2088	961	183	5.85	2.54	1.22	3.42	.29	13.3	.248
.61	889	568	48	7.9R- .064- .0076	2183	999	110	5.56	2.73	1.56	3.42	.29	13.6	.213
.61	978	568	48	7.9R- .064- .0076	2162	1098	120	5.56	3.08	2.10	3.42	.34	14.5	.187
.61	778	1140	48	7.9R- .127- .0076	2143	1008	230	5.85	2.59	1.22	3.47	.29	13.4	.510
.61	889	1140	48	7.9R- .064- .0076	2454	1029	140	5.56	2.83	1.66	3.47	.29	13.8	.425
.61	978	1140	48	11.8R- .064- .0076	2499	1100	122	5.71	3.12	2.10	3.47	.34	14.7	.374
.61	778	114	345	7.9R- .381- .0076	2068	931	153	7.12	10.7	2.05	2.88	.78	23.6	.0510
.61	889	114	345	11.8R- .381- .0076	2069	975	86	8.05	10.4	1.81	2.88	.78	23.9	.0425
.61	978	114	345	11.8R- .381- .0076	2069	1058	80	8.05	11.2	2.34	2.88	.83	25.4	.0374
.61	778	568	345	7.9R- .254- .0076	2072	1122	344	6.49	12.2	2.83	3.42	.78	25.7	.248
.61	889	568	345	15.8R- .127- .0076	2094	1046	157	6.49	13.1	3.56	3.42	.78	27.4	.213
.61	978	568	345	15.8R- .127- .0076	2089	1145	169	6.49	15.1	5.27	3.42	.83	31.1	.187
.61	778	1140	345	11.8R- .254- .0076	2090	1069	291	7.12	12.1	2.54	3.47	.78	26.0	.510
.61	889	1140	345	15.8R- .127- .0076	2183	1052	163	6.49	13.3	3.76	3.47	.78	27.8	.425
.61	978	1140	345	15.8R- .127- .0076	2160	1155	177	6.49	15.3	5.47	3.47	.83	31.5	.374
.61	778	114	689	11.8R- .381- .0076	2069	872	94	8.05	18.5	2.73	2.88	1.12	33.3	.0510
.61	889	114	689	11.8R- .381- .0076	2069	975	86	8.05	18.4	2.69	2.88	1.12	33.2	.0425
.61	978	114	689	15.8R- .381- .0076	2069	1034	56	8.97	19.3	3.12	2.88	1.17	35.5	.0374
.61	778	568	689	15.8R- .381- .0076	2071	1041	263	8.97	19.7	3.27	3.42	1.12	36.5	.248
.61	889	568	689	15.8R- .381- .0076	2072	1137	248	8.97	21.8	4.34	3.42	1.12	39.7	.213
.61	978	568	689	15.8R- .127- .0076	2089	1147	169	6.49	26.9	7.76	3.42	1.17	45.7	.187
.61	778	1140	689	15.8R- .254- .0076	2096	1032	254	7.76	20.9	3.81	3.47	1.12	37.1	.510
.61	889	1140	689	15.8R- .127- .0076	2183	1052	163	6.49	23.8	5.52	3.47	1.12	40.4	.425
.61	978	1140	689	15.8R- .127- .0076	2160	1155	177	6.49	27.2	8.10	3.47	1.17	46.5	.374
1.52	778	114	48	7.9R- .127- .0076	2073	852	74	5.85	2.39	1.07	2.05	.20	11.6	.0510
1.52	889	114	48	7.9R- .127- .0076	2072	969	80	5.85	2.44	1.07	2.05	.20	11.6	.0425
1.52	978	114	48	7.9R- .127- .0076	2072	1056	78	5.85	2.64	1.37	2.05	.20	12.1	.0374
1.52	778	568	48	7.9R- .127- .0076	2174	880	102	5.85	2.44	1.07	2.98	.20	12.6	.248
1.52	889	568	48	7.9R- .127- .0076	2160	993	104	5.85	2.44	1.12	2.98	.20	12.6	.213
1.52	978	568	48	7.9R- .127- .0076	2149	1084	106	5.85	2.73	1.46	2.98	.20	13.2	.187
1.52	778	1140	48	7.9R- .127- .0076	2434	907	129	5.85	2.49	1.12	3.03	.20	12.7	.510
1.52	889	1140	48	7.9R- .127- .0076	2374	1023	134	5.85	2.54	1.17	3.03	.20	12.8	.425
1.52	978	1140	48	7.9R- .127- .0076	2336	1115	137	5.85	2.83	1.61	3.03	.20	13.5	.374

NOTE: Panel width, w = 0.61 m

TABLE 6

MINIMUM WEIGHT CONCEPT 2 PANEL WEIGHTS FOR SELECTED HEATING, LOADING AND COOLANT OUTLET TEMPERATURE CONDITIONS

(a) U.S. CUSTOMARY UNITS

T_{CO}, °R	q/A, Btu/sec-ft²	P, psi	Fin Geometry: N, h_{fin}, t_{fin}, fins/in. in. in.	P_{CI}, psi	T_{DMW}, °F	$T_{DMW}-T_{CO}$ °F	Heat Exchanger wt, lb/ft²	Prime Panel wt, lb/ft²	Beam wt, lb/ft²	Clip wt, lb/ft²	Manifold wt, lb/ft²	Seal wt, lb/ft²	Total wt, lb/ft²	Methane Rate, lb/sec-ft²
1400	10	6.95	20R0-0.025-0.003	305	989	49	0.72	0.93	0.38	0.11	0.59	0.06	2.79	0.01045
1600	10	6.95	20R0- .025- .003	305	1189	49	.72	.93	.39	.12	.59	.05	2.81	.00872
1760	10	6.95	20R0- .025- .003	304	1348	48	.72	.91	.49	.15	.59	.07	2.93	.00766
1400	50	6.95	20R0- .025- .003	388	1052	112	.72	.93	.38	.11	.70	.06	2.90	.0509
1600	50	6.95	20R0- .025- .003	374	1250	110	.72	.93	.39	.12	.70	.06	2.92	.0436
1760	50	6.95	20R0- .025- .003	364	1409	109	.72	.91	.49	.15	.70	.07	3.04	.0383
1400	100	6.95	20R0- .025- .003	580	1096	156	.72	.93	.38	.11	.71	.06	2.91	.1045
1600	100	6.95	20R0- .025- .003	532	1296	156	.72	.93	.39	.12	.71	.06	2.93	.0872
1760	100	6.95	30R0- .025- .003	545	1433	133	.75	.91	.49	.15	.71	.07	3.08	.0766
1400	10	50	20R0- .025- .003	305	989	49	.72	.95	.94	.34	.59	.16	4.70	.01045
1600	10	50	20R0- .025- .003	305	1189	49	.72	.95	2.07	.37	.59	.16	4.86	.00872
1760	10	50	20R0- .025- .003	304	1348	48	.72	.00	2.26	.41	.59	.17	5.15	.00766
1400	50	50	20R0- .025- .003	388	1052	112	.72	.95	1.94	.34	.70	.16	4.81	.0509
1600	50	50	20R0- .025- .003	374	1250	110	.72	.95	2.07	.37	.70	.16	4.97	.0436
1760	50	50	20R0- .025- .003	364	1409	109	.72	1.00	2.26	.41	.70	.17	5.26	.0383
1400	100	50	20R0- .025- .003	580	1096	156	.72	.95	1.94	.34	.72	.16	4.83	.1045
1600	100	50	20R0- .025- .003	532	1296	156	.72	.95	2.07	.37	.73	.16	5.00	.0872
1760	100	50	30R0- .025- .003	545	1433	133	.75	1.00	2.26	.41	.77	.17	5.36	.0766
1400	250	50	20R0- .038- .003	716	1076	136	.75	.95	1.94	.34	.85	.16	4.99	.2545
1600	250	50	30R0- .050- .003	563	1276	136	.85	.95	2.07	.37	.86	.16	5.26	.218
1760	250	50	No design	-	-	-	-	1.00	2.26	.41	.90	.17	-	.1915
1400	500	50	40R0- .072- .003	830	1046	106	1.03	.95	1.94	.34	.95	.16	5.37	.509
1600	500	50	No design	-	-	-	-	.95	2.07	.37	.96	.16	-	.436
1760	500	50	No design	-	-	-	-	1.00	2.26	.41	1.00	.17	-	.383
1400	10	100	20R0- .025- .003	305	989	49	.72	1.08	3.10	.37	.59	.23	6.09	.01045
1600	10	100	20R0- .025- .003	305	1189	49	.72	1.11	3.19	.37	.59	.23	6.21	.00872
1760	10	100	20R0- .025- .003	304	1348	48	.72	1.18	3.51	.41	.59	.24	6.65	.00766
1400	50	100	20R0- .025- .003	388	1052	112	.72	1.08	3.10	.37	.70	.23	6.20	.0509
1600	50	100	20R0- .025- .003	374	1250	110	.72	1.11	3.19	.37	.70	.23	6.32	.0436
1760	50	100	20R0- .025- .003	364	1409	109	.72	1.18	3.51	.41	.70	.24	6.76	.0383
1400	100	100	20R0- .025- .003	580	1096	156	.72	1.08	3.10	.37	.72	.23	6.22	.1045
1600	100	100	20R0- .025- .003	532	1296	156	.72	1.11	3.19	.37	.73	.23	6.35	.0872
1760	100	100	30R0- .025- .003	545	1433	133	.75	1.18	3.51	.41	.77	.24	6.86	.0766
1400	250	100	20R0- .038- .003	716	1076	136	.75	1.08	3.10	.37	.85	.23	6.38	.2545
1600	250	100	30R0- .050- .003	563	1276	136	.85	1.11	3.19	.37	.86	.23	6.61	.218
1760	250	100	No design	-	-	-	-	1.18	3.51	.41	.90	.24	-	.1915
1400	500	100	40R0- .072- .003	830	1046	106	1.03	1.08	3.10	.37	.95	.23	6.76	.509
1600	500	100	No design	-	-	-	-	1.11	3.19	.37	.96	.23	-	.436
1760	500	100	No design	-	-	-	-	1.18	3.51	.41	1.00	.24	-	.383
1400	10	250	20R0- .025- .003	305	989	49	.72	1.86	5.49	.38	.59	.36	9.40	.01045
1600	10	250	20R0- .025- .003	305	1189	49	.72	1.93	5.66	.38	.59	.36	9.64	.00872
1760	10	250	20R0- .025- .003	304	1348	48	.72	2.12	6.26	.42	.59	.38	10.49	.00766
1400	50	250	20R0- .025- .003	388	1052	112	.72	1.86	5.49	.38	.70	.36	9.51	.0509
1600	50	250	20R0- .025- .003	374	1250	110	.72	1.93	5.66	.38	.70	.36	9.75	.0436
1760	50	250	20R0- .025- .003	364	1409	109	.72	2.12	6.26	.42	.70	.38	10.60	.0383
1400	100	250	20R0- .025- .003	580	1096	156	.72	1.86	5.49	.38	.72	.36	9.53	.1045
1600	100	250	20R0- .025- .003	532	1296	156	.72	1.93	5.66	.38	.73	.36	9.78	.0872
1760	100	250	30R0- .025- .003	545	1433	133	.75	2.12	6.26	.42	.77	.38	10.70	.0766
1400	250	250	20R0- .038- .003	716	1076	136	.75	1.86	5.49	.38	.85	.36	9.69	.2545
1600	250	250	30R0- .050- .003	563	1276	136	.85	1.93	5.66	.38	.86	.36	10.04	.218
1760	250	250	No design	-	-	-	-	2.12	6.26	.42	.90	.38	-	.1915
1400	500	250	40R0- .072- .003	830	1046	106	1.03	1.86	5.49	.38	.95	.36	10.07	.509
1600	500	250	No design	-	-	-	-	1.93	5.66	.38	.96	.36	-	.436
1760	500	250	No design	-	-	-	-	2.12	6.26	.42	1.00	.38	-	.383

NOTES: (1) Panel width, w = 2 ft
(2) Panel length, ℓ = 2 ft

TABLE 6 (Concluded)

(b) SI UNITS

T_{CO}, °K	q/A, kW/m²	P, kN/m²	Fin Geometry N, h_{fin}, t_{fin}, fins/cm cm cm	P_{CI}, kN/m²	T_{DHW}, °K	$T_{DHW}-T_{CO}$ °K	Heat Exchanger, wt, kg/m²	Prime Panel wt, kg/m²	Beam wt, kg/m²	Clip wt, kg/m²	Manifold wt, kg/m²	Seal wt, kg/m²	Total wt, kg/m²	Methane rate, kg/s-m²
778	114	48	7.9R0-0.064-0.0076	2100	805	27	3.52	4.54	1.86	0.54	2.88	0.29	13.5	0.0510
889	114	48	7.9R0- .064- .0076	2100	916	27	3.52	4.54	1.91	.59	2.88	.29	13.7	.0425
978	114	48	7.9R0- .064- .0076	2090	1005	27	3.52	4.44	2.39	.73	2.88	.34	14.3	.0374
778	568	48	7.9R0- .064- .0076	2670	840	62	3.52	4.54	1.86	.54	3.42	.29	14.2	.248
889	568	48	7.9R0- .064- .0076	2580	950	61	3.52	4.54	1.91	.59	3.42	.29	14.3	.213
978	568	48	7.9R0- .064- .0076	2510	1039	61	3.52	4.44	2.39	.73	3.42	.34	14.8	.187
778	1140	48	7.9R0- .064- .0076	4000	865	87	3.52	4.54	1.86	.54	3.47	.29	14.2	.510
889	1140	48	7.9R0- .064- .0076	3660	976	87	3.52	4.54	1.91	.59	3.47	.29	14.3	.425
978	1140	48	11.8R0- .064- .0076	3760	1052	74	3.66	4.44	2.39	.73	3.47	.34	15.0	.374
778	114	345	7.9R0- .064- .0076	2100	805	27	3.52	4.64	9.49	1.66	2.88	.78	23.0	.0510
889	114	345	7.9R0- .064- .0076	2100	916	27	3.52	4.64	10.10	1.81	2.88	.78	23.7	.0425
978	114	345	7.9R0- .064- .0076	2090	1005	27	3.52	4.88	11.05	2.00	2.88	.83	25.1	.0374
778	568	345	7.9R0- .064- .0076	2670	840	62	3.52	4.64	9.49	1.66	3.42	.78	23.5	.248
889	568	345	7.9R0- .064- .0076	2580	950	61	3.52	4.64	10.10	1.81	3.42	.78	24.3	.213
978	568	345	7.9R0- .064- .0076	2510	1039	61	3.52	4.88	11.05	2.00	3.42	.83	25.7	.187
778	1140	345	7.9R0- .064- .0076	4000	865	87	3.52	4.64	9.49	1.66	3.52	.78	23.6	.510
889	1140	345	7.9R0- .064- .0076	3660	976	87	3.52	4.64	10.10	1.81	3.56	.78	24.4	.425
978	1140	345	11.8R0- .064- .0076	3760	1052	74	3.66	4.88	11.05	2.00	3.76	.83	26.2	.374
778	2840	345	7.9R0- .096- .0076	4830	854	76	3.66	4.64	9.49	1.66	4.15	.78	24.4	1.242
889	2840	345	11.8R0- .127- .0076	3880	965	76	4.15	4.64	10.10	1.81	4.20	.78	25.7	1.065
978	2840	345	No design	-	-	-	-	4.88	11.05	2.00	4.39	.83	-	.935
778	5680	345	15.8R0- .183- .0076	5710	837	59	5.03	4.64	9.49	1.66	4.64	.78	26.2	2.48
889	5680	345	No design	-	-	-	-	4.64	10.10	1.81	4.69	.78	-	2.13
978	5680	345	No design	-	-	-	-	4.88	11.05	2.00	4.88	.83	-	1.87
778	114	689	7.9R0- .064- .0076	2100	805	27	3.52	5.28	15.15	1.81	2.88	1.12	29.7	.0510
889	114	689	7.9R0- .064- .0076	2100	916	27	3.52	5.42	15.60	1.81	2.88	1.12	30.3	.0425
978	114	689	7.9R0- .064- .0076	2090	1005	27	3.52	5.76	17.15	2.00	2.88	1.17	32.5	.0374
778	568	689	7.9R0- .064- .0076	2670	840	62	3.52	5.28	15.15	1.81	3.42	1.12	30.3	.248
889	568	689	7.9R0- .064- .0076	2580	950	61	3.52	5.42	15.60	1.81	3.42	1.12	30.9	.213
978	568	689	7.9R0- .064- .0076	2510	1039	61	3.52	5.76	17.15	2.00	3.42	1.17	33.0	.187
778	1140	689	7.9R0- .064- .0076	4000	865	87	3.52	5.28	15.15	1.81	3.52	1.12	30.4	.510
889	1140	689	7.9R0- .064- .0076	3660	976	87	3.52	5.42	15.60	1.81	3.56	1.12	31.0	.425
978	1140	689	11.8R0- .064- .0076	3760	1052	74	3.66	5.76	17.15	2.00	3.76	1.17	33.5	.374
778	2840	689	7.9R0- .096- .0076	4830	854	76	3.66	5.28	15.15	1.81	4.15	1.12	31.2	1.242
889	2840	689	11.8R0- .127- .0076	3880	965	76	4.15	5.42	15.60	1.81	4.20	1.12	32.3	1.065
978	2840	689	No design	-	-	-	-	5.76	17.15	2.00	4.39	1.17	-	.935
778	5680	689	15.8R0- .183- .0076	5710	837	59	5.03	5.28	15.15	1.81	4.64	1.12	33.0	2.48
889	5680	689	No design	-	-	-	-	5.42	15.60	1.81	4.69	1.12	-	2.13
978	5680	689	No design	-	-	-	-	5.76	17.15	2.00	4.88	1.17	-	1.87
778	114	1720	7.9R0- .064- .0076	2100	805	27	3.52	9.08	26.80	1.86	2.88	1.76	45.9	.0510
889	114	1720	7.9R0- .064- .0076	2100	916	27	3.52	9.41	27.60	1.86	2.88	1.76	47.1	.0425
978	114	1720	7.9R0- .064- .0076	2090	1005	27	3.52	10.35	30.60	2.05	2.88	1.86	51.2	.0374
778	568	1720	7.9R0- .064- .0076	2670	840	62	3.52	9.08	26.80	1.86	3.42	1.76	46.4	.248
889	568	1720	7.9R0- .064- .0076	2580	950	61	3.52	9.41	27.60	1.86	3.42	1.76	47.6	.213
978	568	1720	7.9R0- .064- .0076	2510	1039	61	3.52	10.35	30.60	2.05	3.42	1.86	51.8	.187
778	1140	1720	7.9R0- .064- .0076	4000	865	87	3.52	9.08	26.80	1.86	3.52	1.76	46.5	.510
889	1140	1720	7.9R0- .064- .0076	3660	976	87	3.52	9.41	27.60	1.86	3.56	1.76	47.8	.425
978	1140	1720	11.8R0- .064- .0076	3760	1052	74	3.66	10.35	30.60	2.05	3.76	1.86	52.2	.374
778	2840	1720	7.9R0- .096- .0076	4830	854	76	3.66	9.08	26.80	1.86	4.15	1.76	47.3	1.242
889	2840	1720	11.8R0- .127- .0076	3880	965	76	4.15	9.41	27.60	1.86	4.20	1.76	49.0	1.065
978	2840	1720	No design	-	-	-	-	10.35	30.60	2.05	4.39	1.86	-	.935
778	5680	1720	15.8R0- .183- .0076	5710	837	59	5.03	9.08	26.80	1.86	4.64	1.76	49.2	2.48
889	5680	1720	No design	-	-	-	-	9.41	27.60	1.86	4.69	1.76	-	2.13
978	5680	1720	No design	-	-	-	-	10.35	30.60	2.05	4.88	1.86	-	1.87

NOTES: (1) Panel width, w = 0.61 cm
(2) Panel length, ℓ = 0.61 cm

Liquid at 300°R	References: Liquid	References: Gas	Gas property on curve	Multiply by	To obtain
-155 Btu/lb	10	6	Enthalpy, $H \times 10^{-2}$, Btu/lb	2.33	J/g
1.7×10^{-5} Btu/sec-°R-ft	10	5, 10	Thermal conductivity, $k \times 10^{5}$, Btu/sec-ft·°R	6230	W m-°K
37×10^{-6} lb/sec-ft	10 extrap	5, 10	Viscosity, $\mu \times 10^{6}$, lb/ft-sec	1.49	N-s/m²
2.28			Prandtl number, Pr × 10	-	-
1.05 Btu/lb-°R	10	5, 10	Specific heat, Cp, Btu/lb °R	4.18	J/g-°K

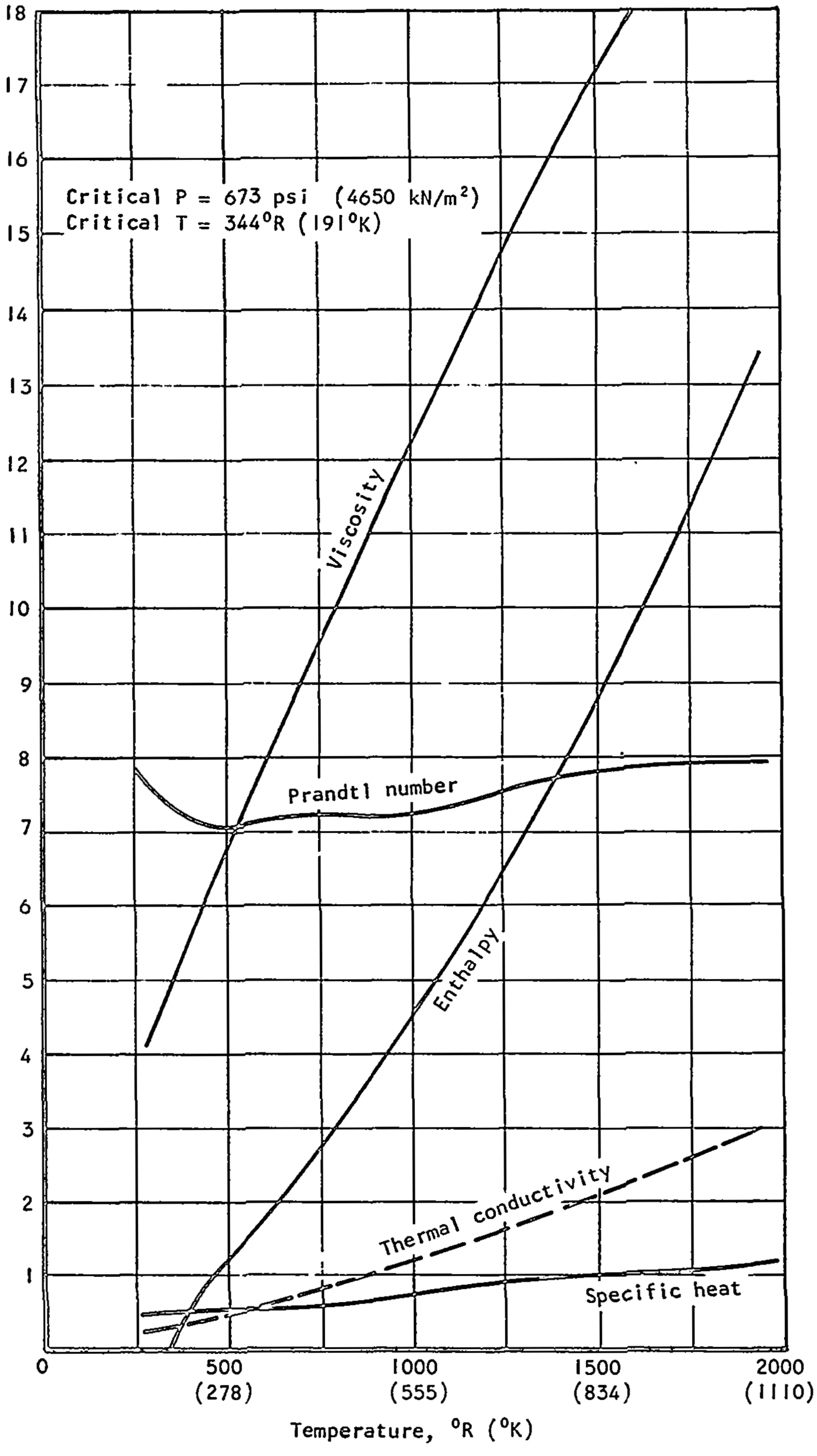

Figure 1. Transport Properties of Methane Gas and Liquid

Property	Multiply by	To obtain
Enthalpy, H x 10^{-3}, Btu/lb	2.33	J/g
Thermal conductivity, k x 10^5, Btu/sec-ft-°R	6230	W/m-°K
Viscosity, μ x 10^6, lb/ft sec	1.49	N-s/m^2
Prandtl number, Pr x 10	-	-
Specific heat, Cp, Btu/lb-°R	4.18	J/g-°K

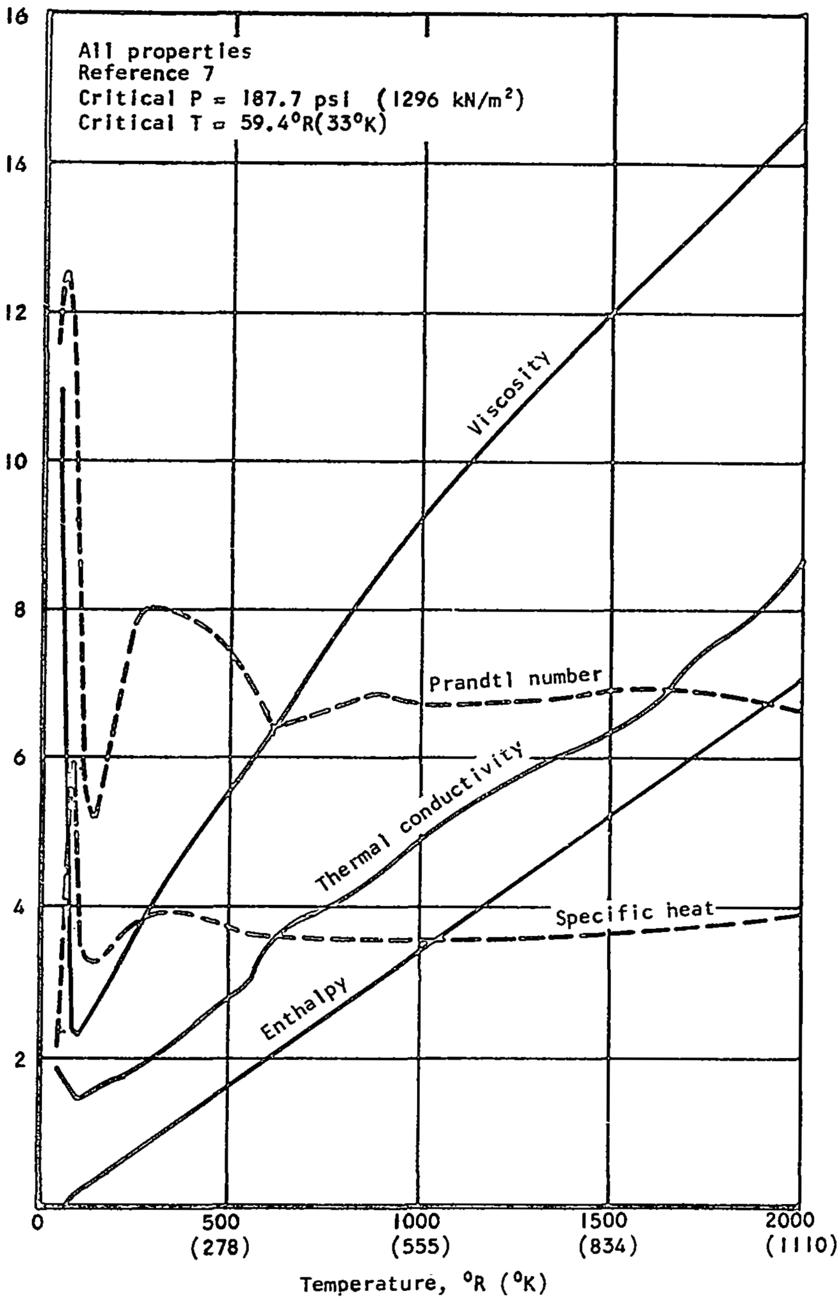

Figure 2. Transport Properties of Para-Hydrogen Gas

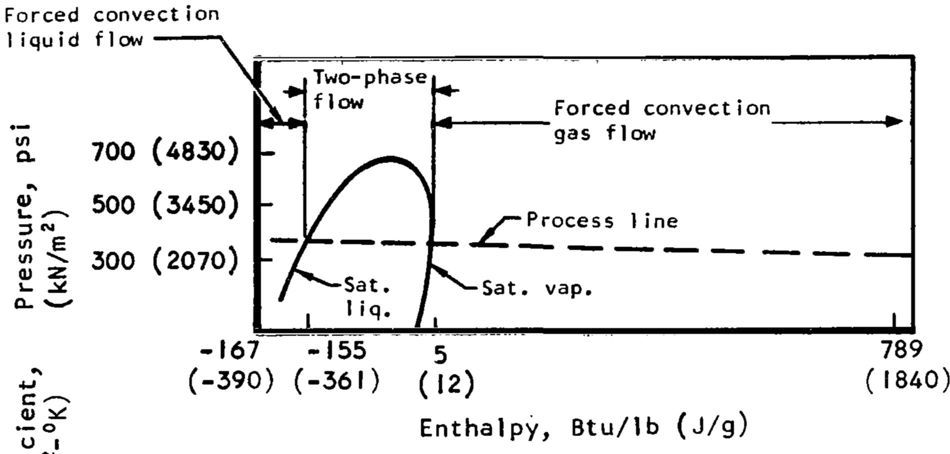

Figure 3a

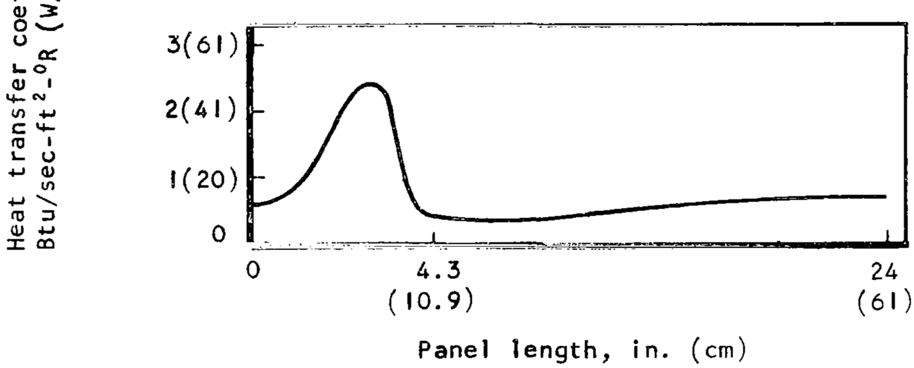

Figure 3b

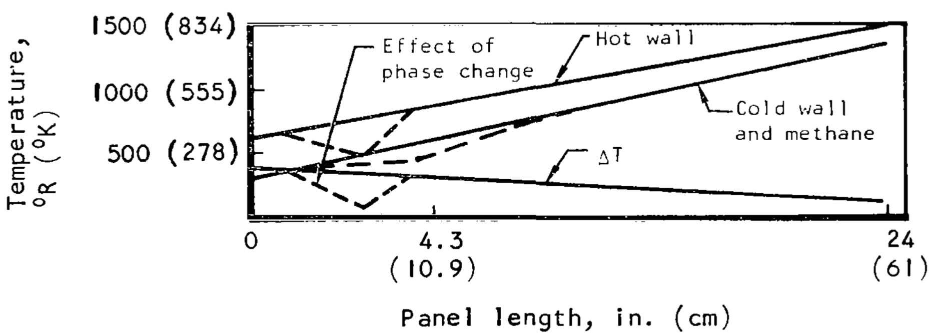

Figure 3c

Figure 3. Typical Effects of Methane Phase Change on Panel with Heat Flux of 100 Btu/sec-ft^2 (1140 kW/m^2) and Offset Fin Geometry, 20(7.9)R0-0.025(0.063)-0.003(0.0076)

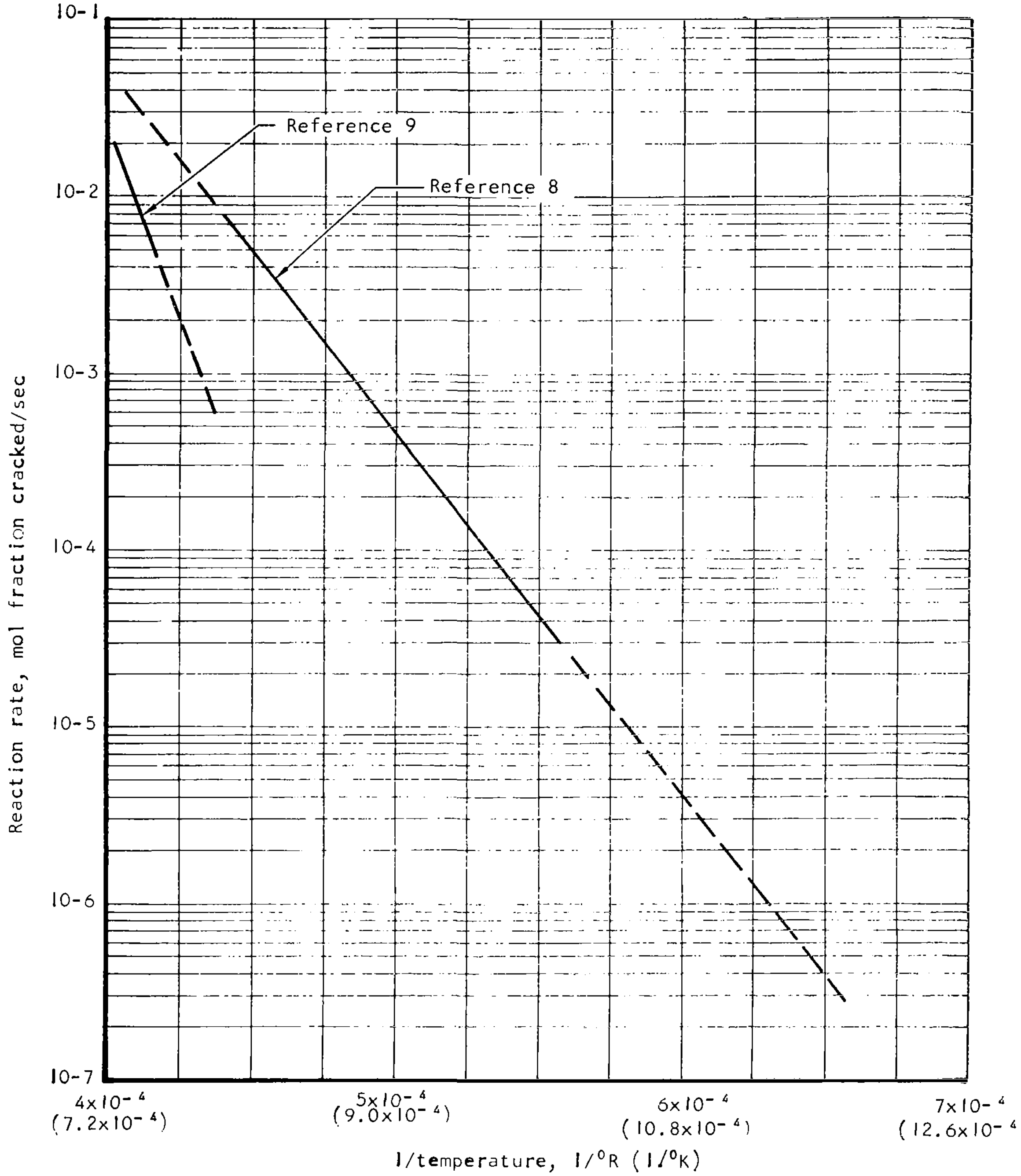

Figure 4. Cracking Reaction Rate of Methane

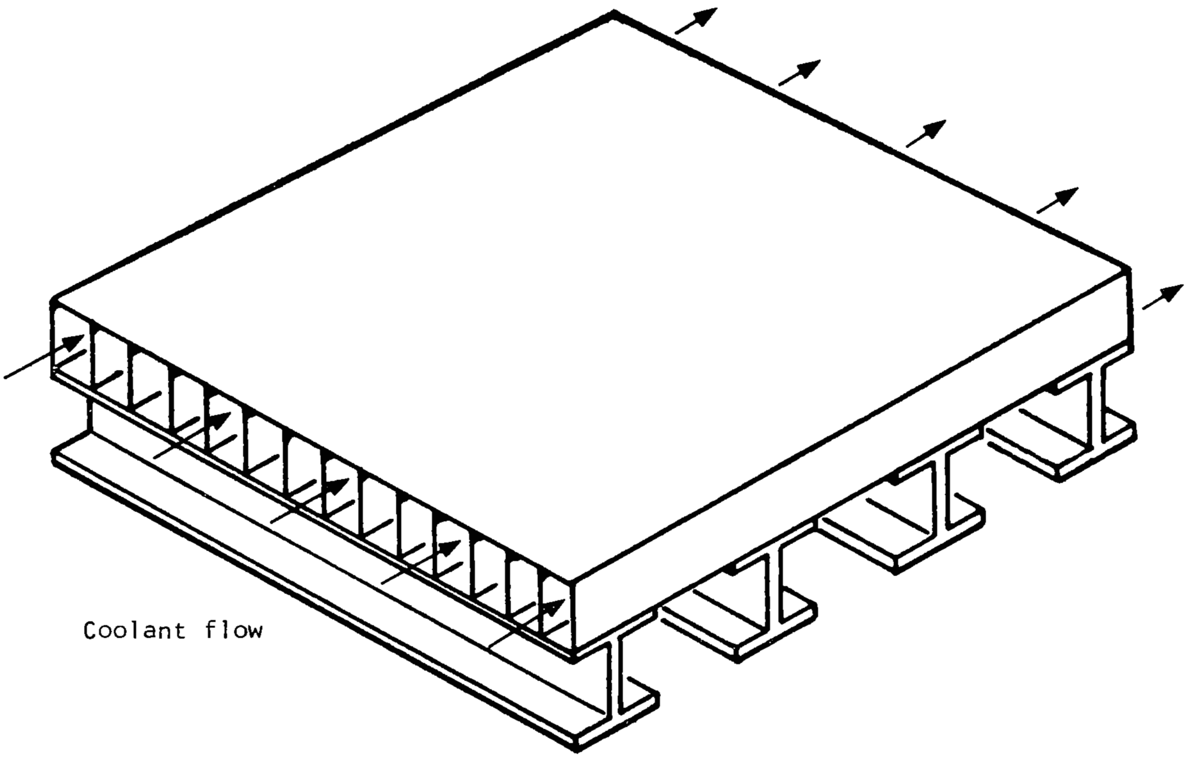

Figure 5. Single Layered Sandwich Panel (Concept 1)

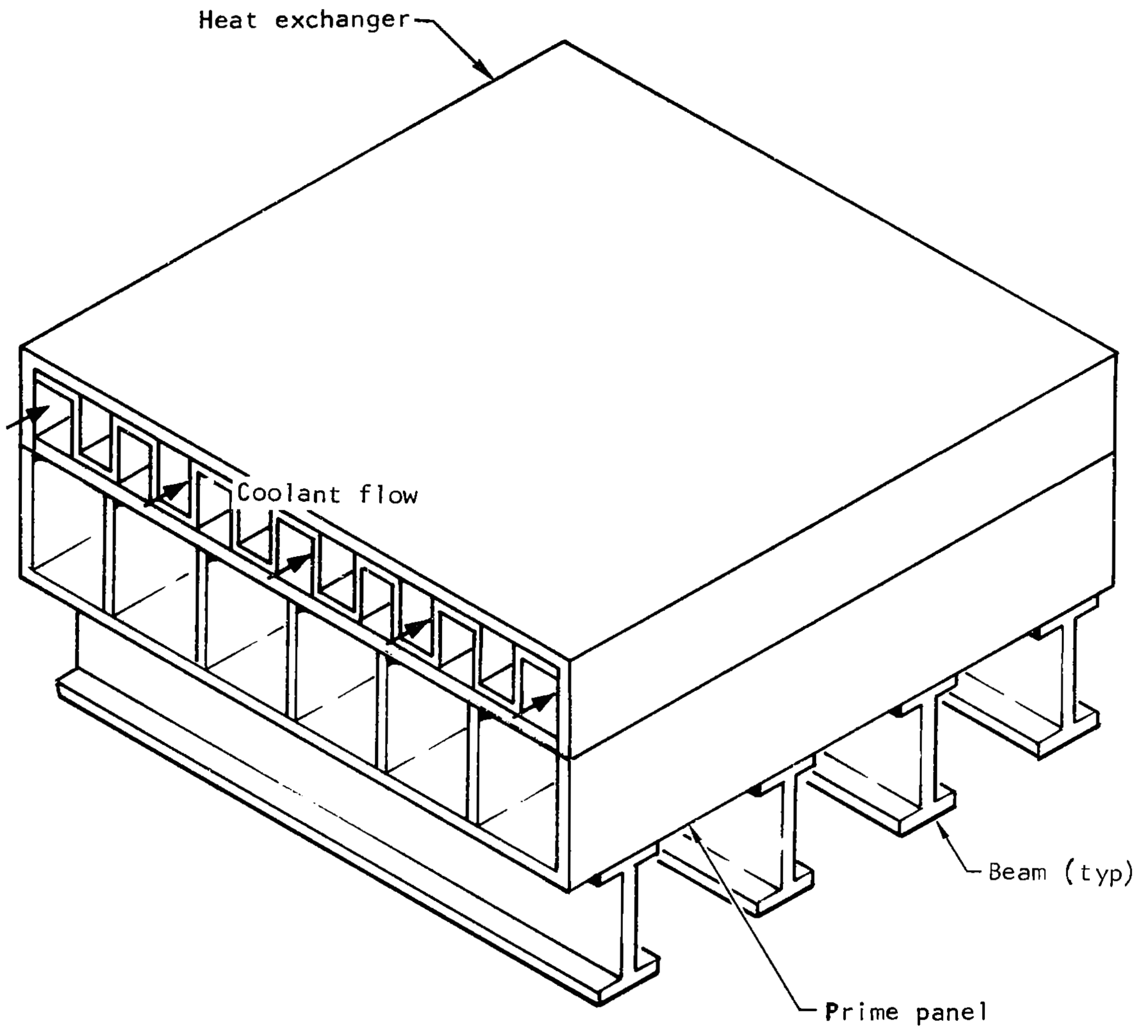

Figure 6. Heat Exchanger Bonded to Prime Panel on I-Beams with Single-Pass Flow (Concept 2)

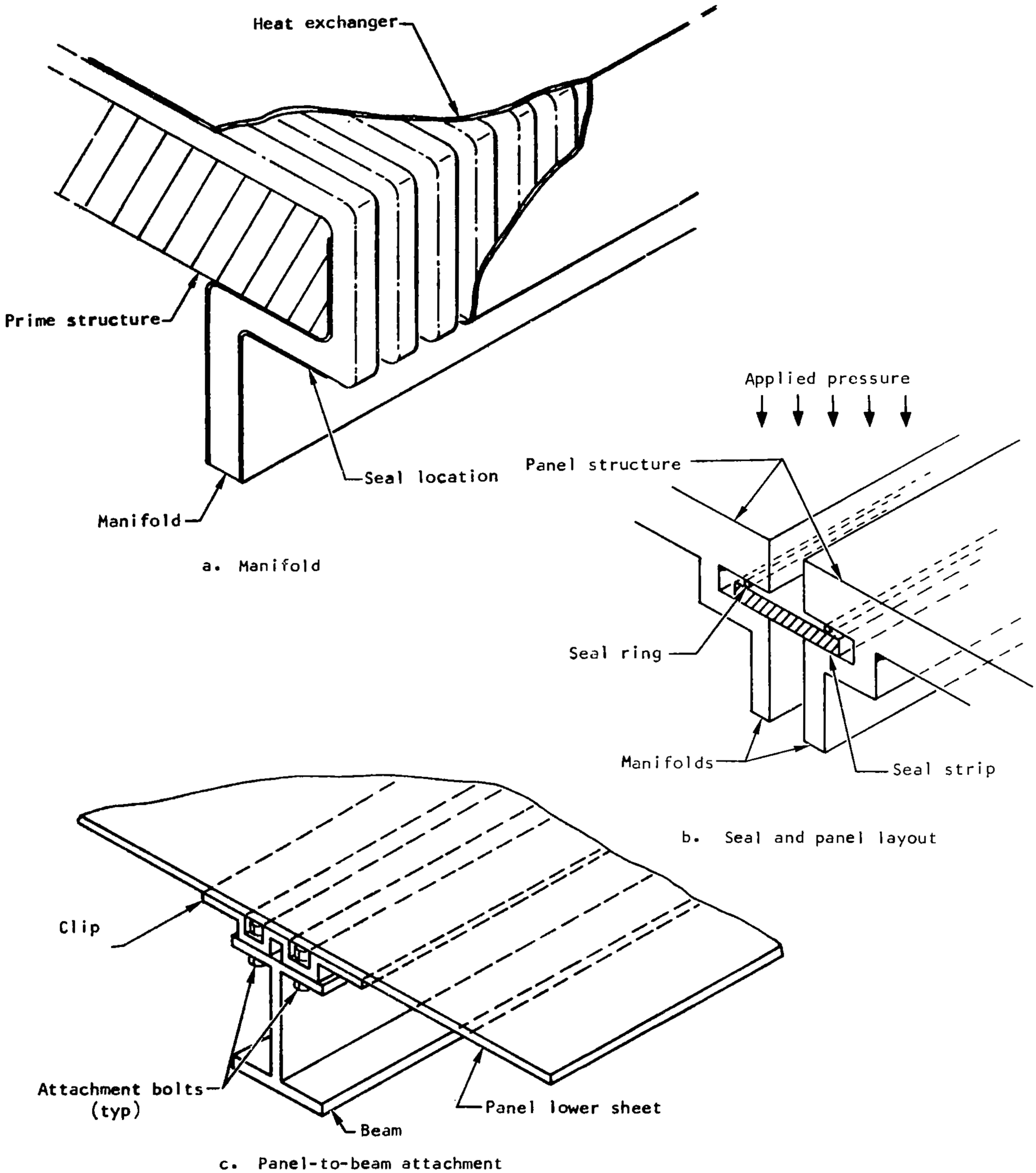

Figure 7. Panel Accessories

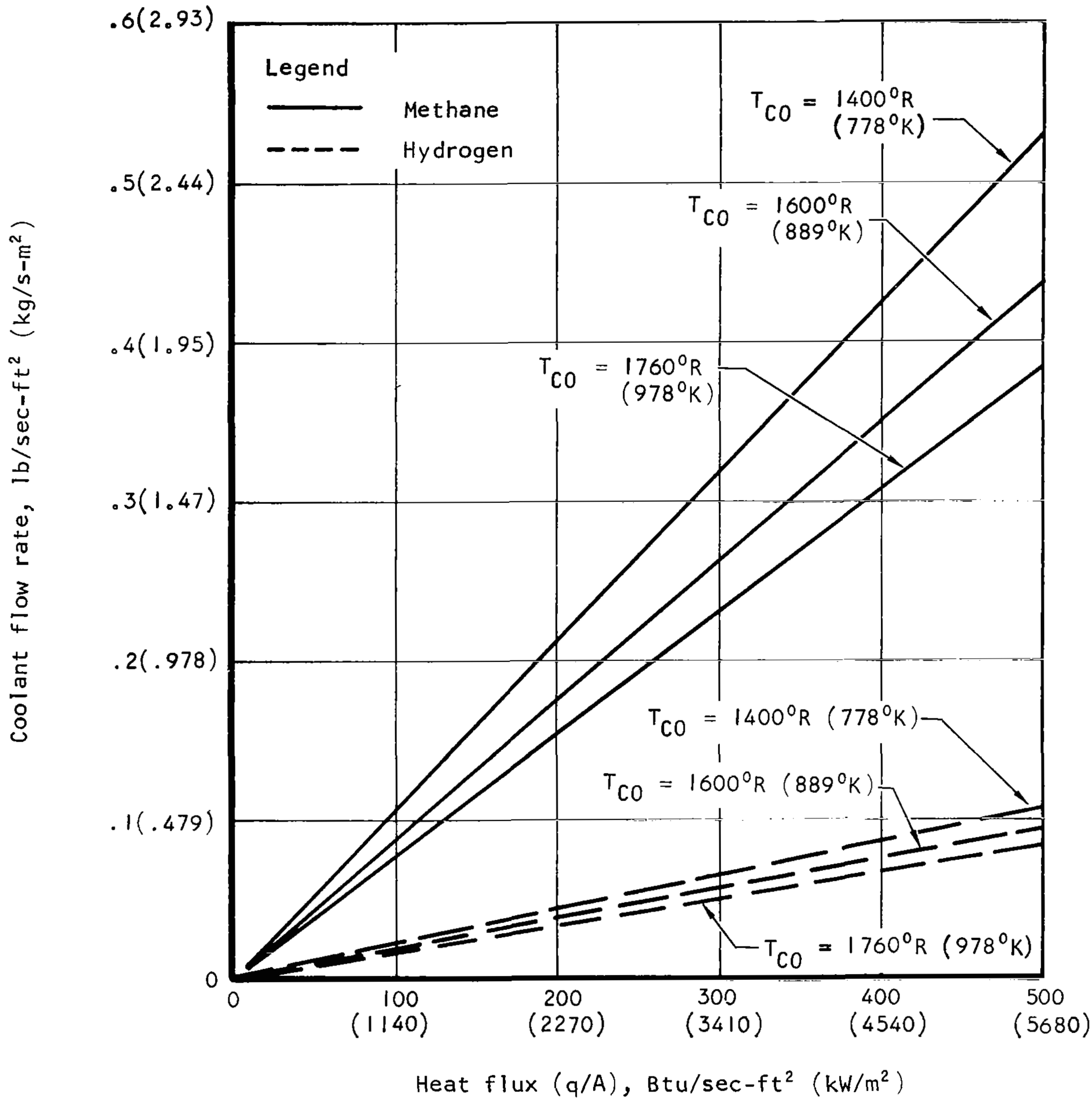

Figure 8. Variation of Coolant Mass Flow Requirements With Heat Flux for Three Coolant Outlet Temperatures

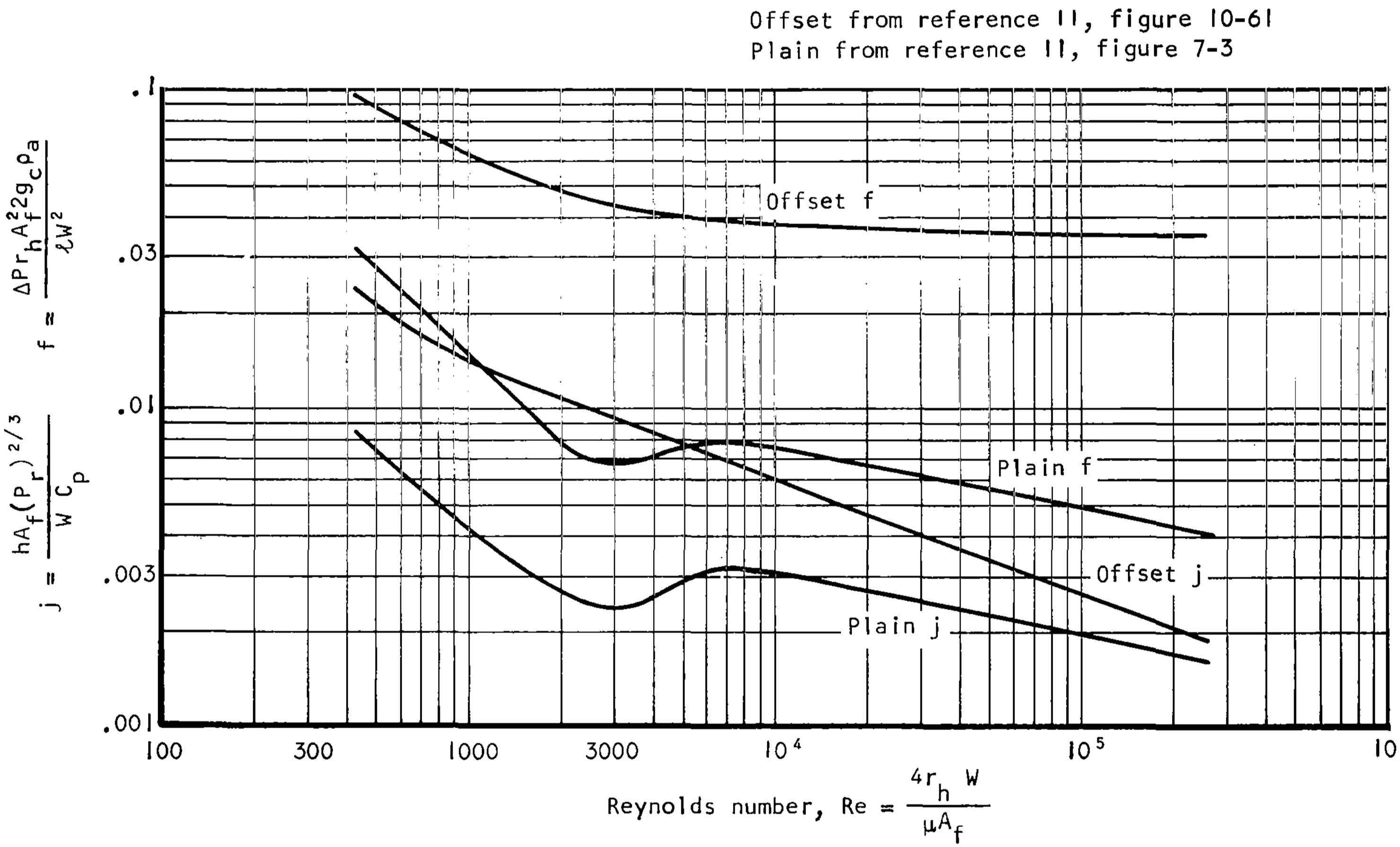

Figure 9. Plain and Offset Fin f and j vs Reynolds Number

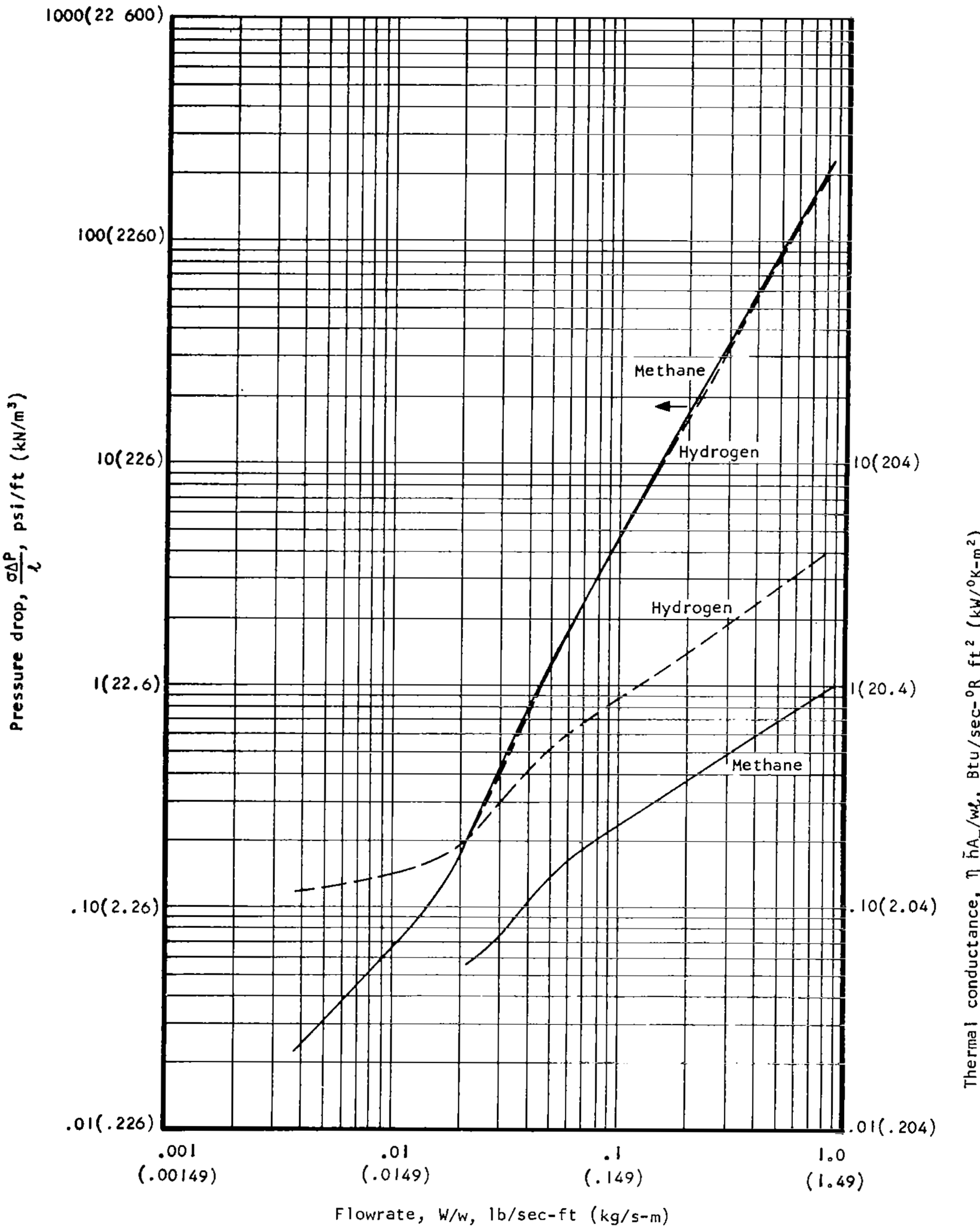

Figure 10. Comparison of Methane and Hydrogen Thermal Conductance with Plain Fin Geometry, 20(7.9) RO-0.025 (0.063)-0.003(0.0076), and 1050°R (583°K) Coolant Temperature

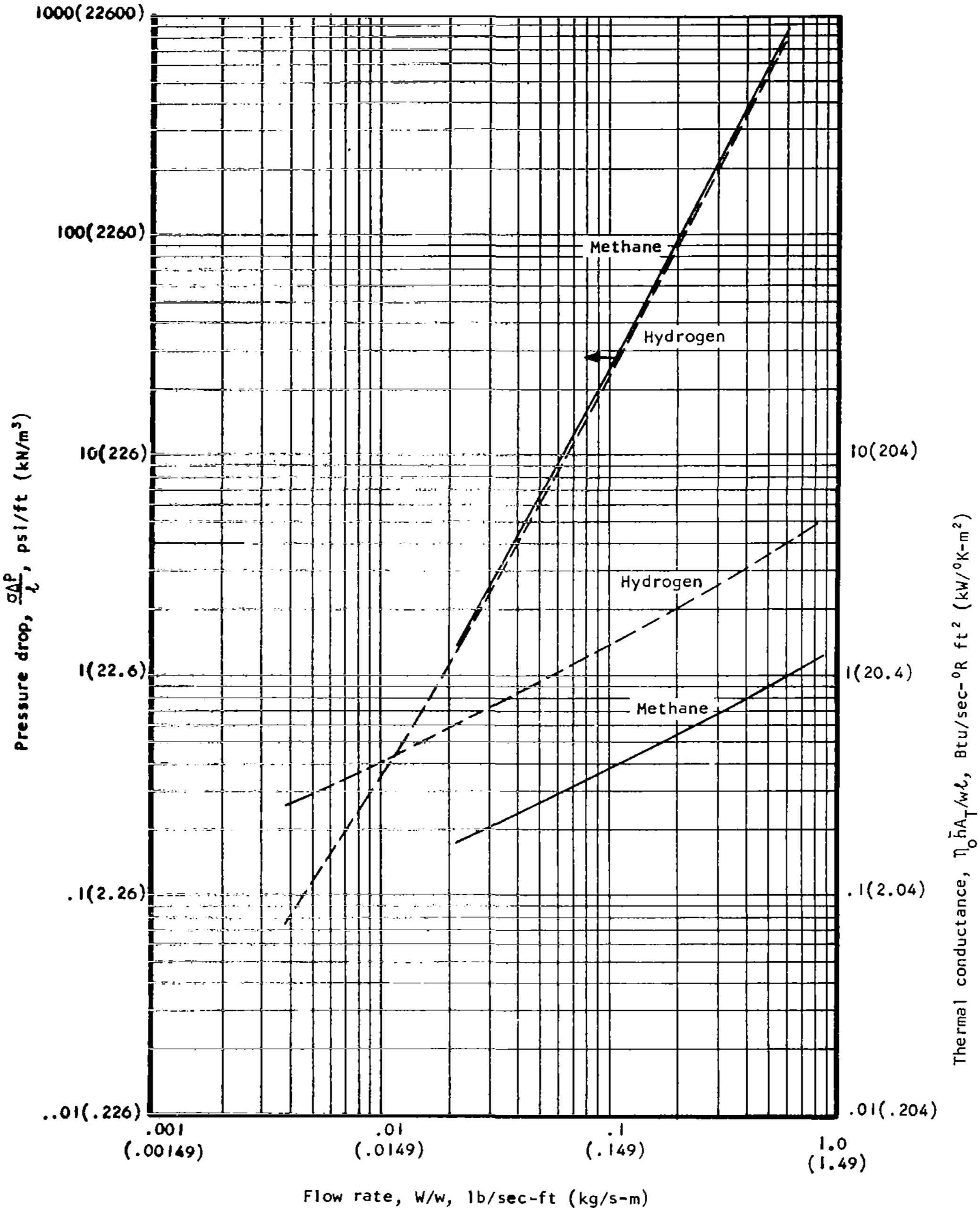

Figure 11. Comparison of Hydrogen and Methane Pressure Drop and Thermal Conductance with Offset Fin Geometry, 20(7.9)R0-0.025(0.063)-0.003(0.0076), and 1050°R (583°K) Coolant Temperature

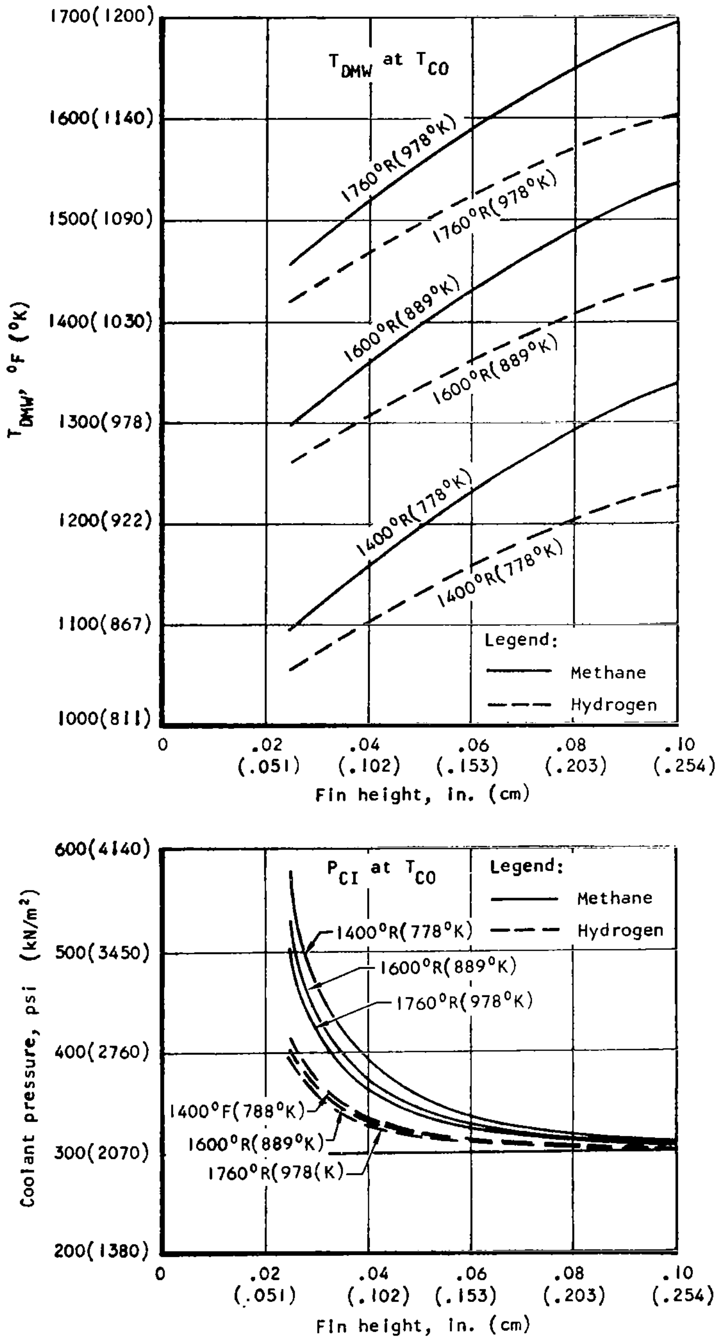

Figure 12. Effect of Fin Height on P_{CI} and T_{DMW} at 100 Btu/sec-ft² (1140 kW/m²) with Offset Fin Geometry, 20(7.9)R0-0.025 (0.063)-0.003(0.0076)

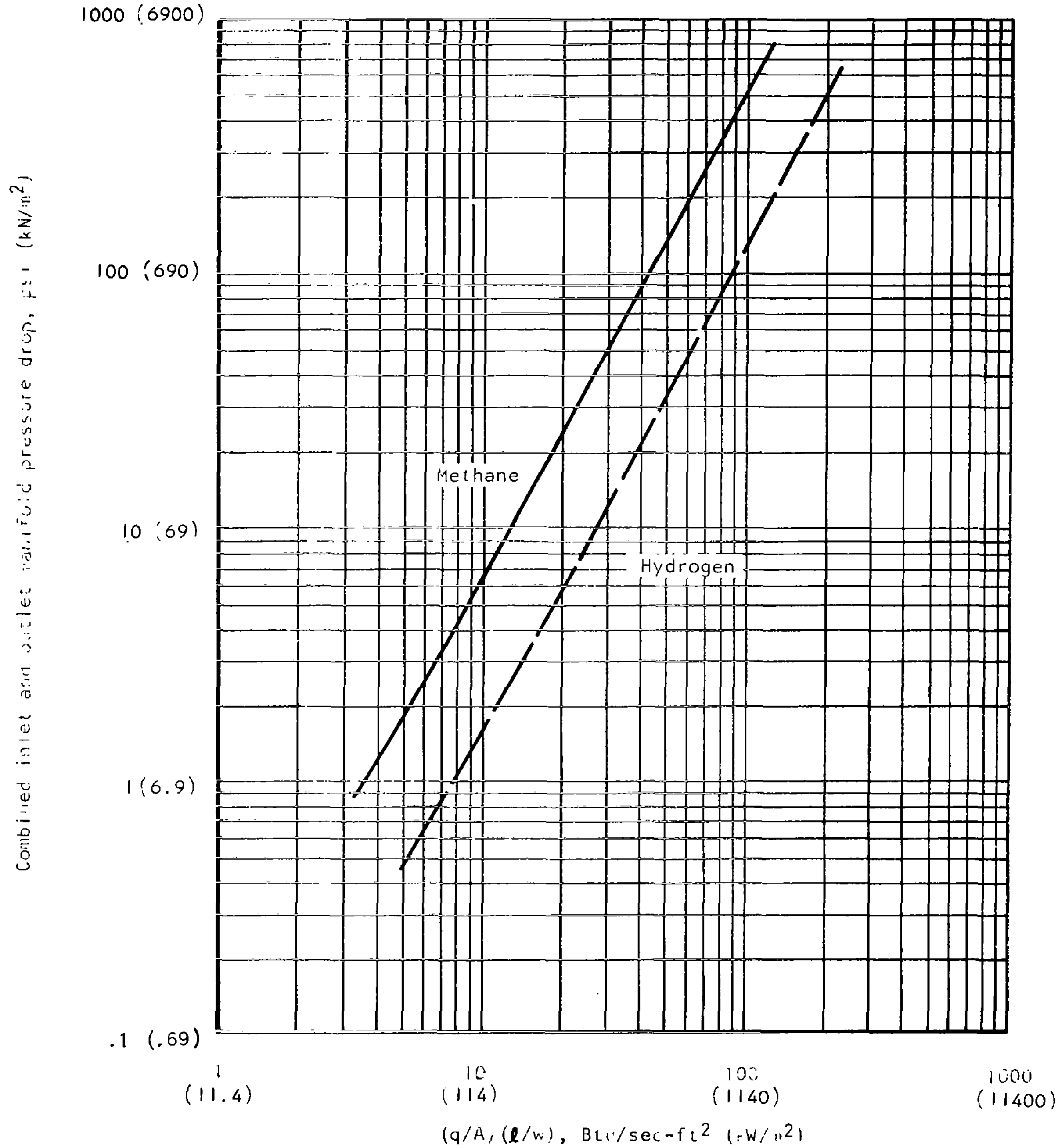

Figure 13. Concept I Outlet Manifold Pressure Drop Comparison of Hydrogen and Methane with T_{CO} 1600°R (889°K), Manifold Width of 24 in. (61 cm), Port Diameter of 1.75 in. (4.45 cm) and 10(3.9)R-0.050(0.127)-0.004 (0.010) Fin Geometry

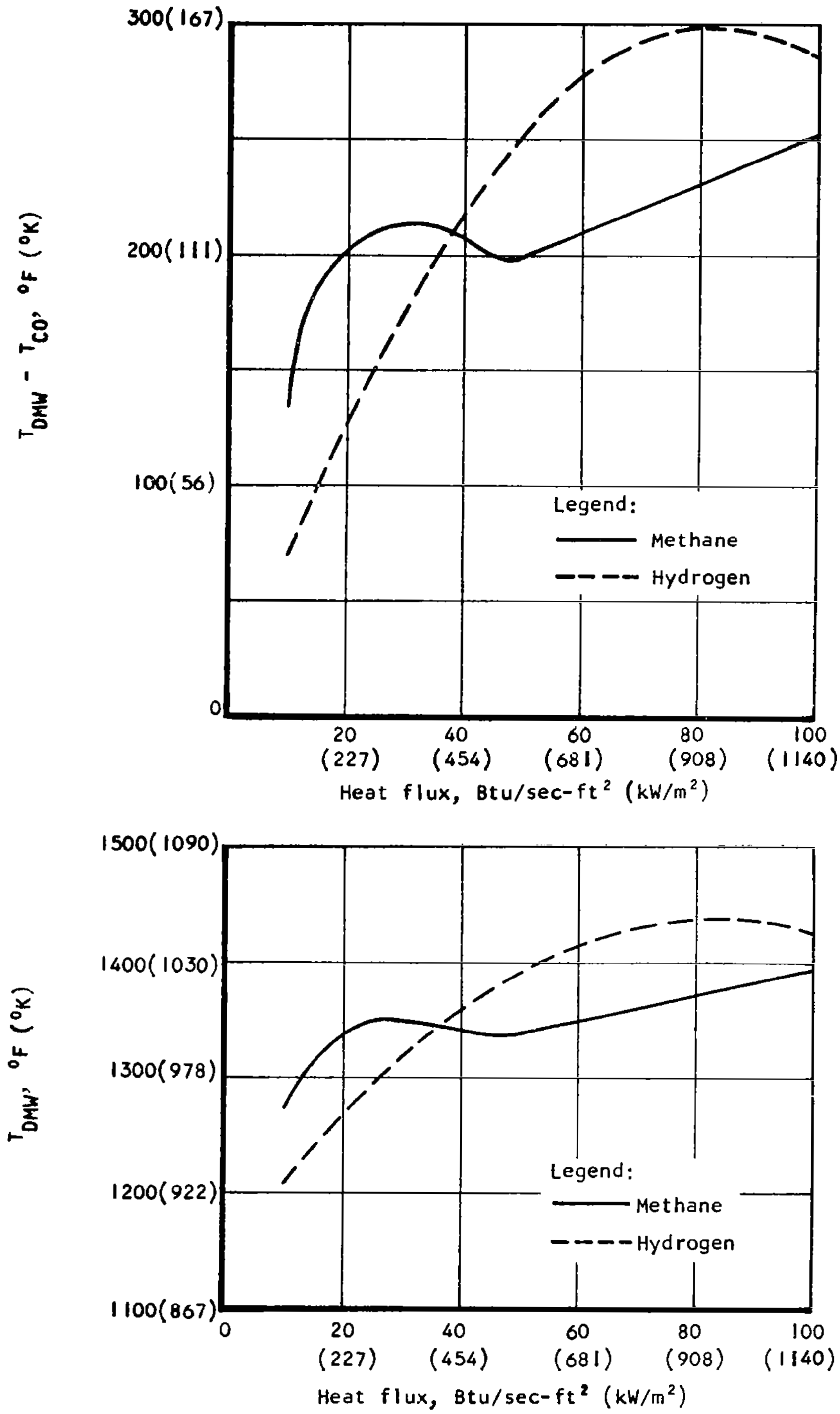

Figure 14. Effect of Heat Flux on Temperature Difference and Wall Temperature with T_{CO} = 1600°R(889°K) and Plain Fin Geometry, 20(7.9)R-0.025(0.063)-0.003(0.0076)

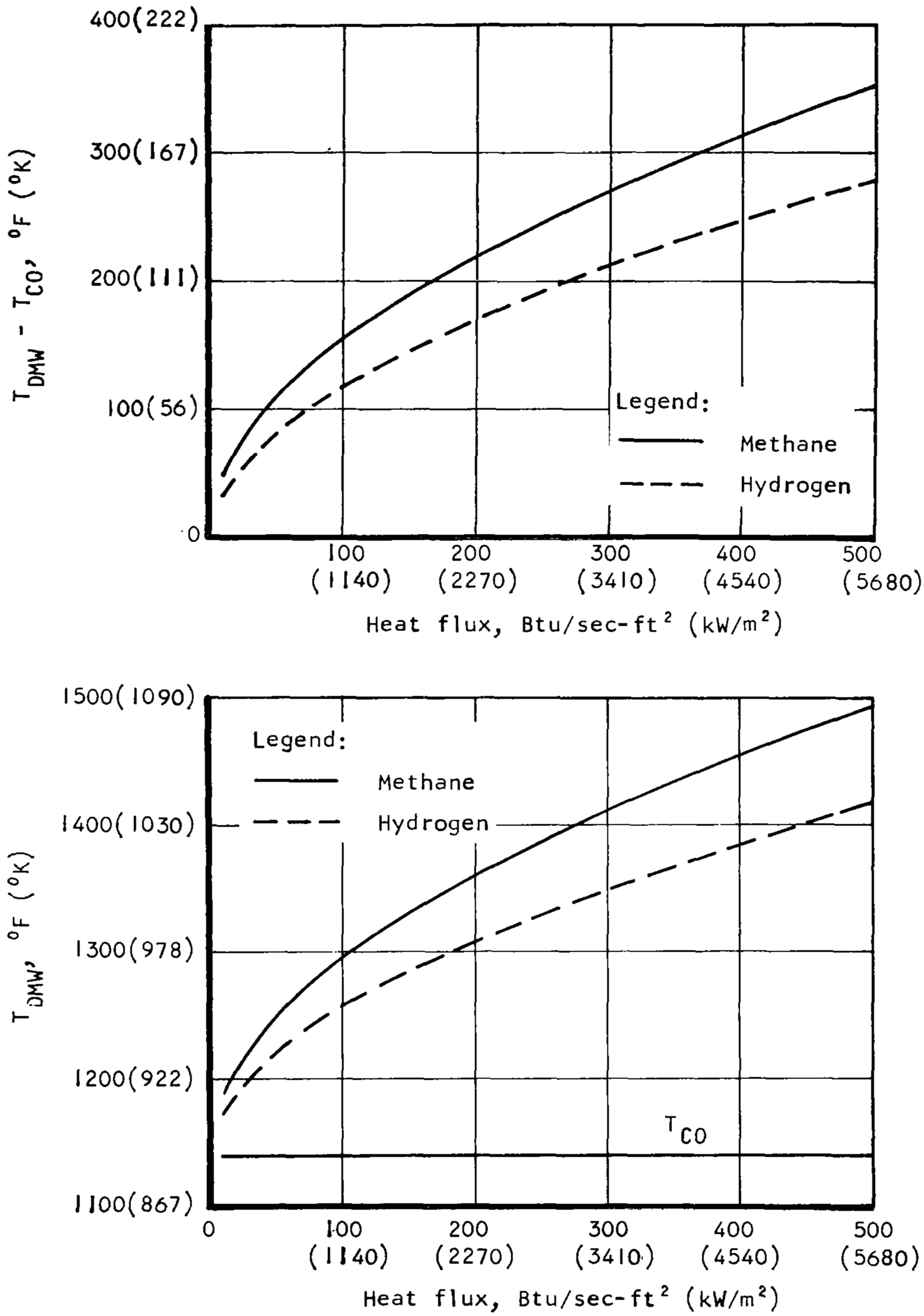

Figure 15. Effect of Heat Flux on Temperature Difference and Wall Temperature with T_{CO} = 1600°R (889°K) and Offset Fin Geometry 20(7.9)R0-0.025(0.063)-0.003(0.0076)

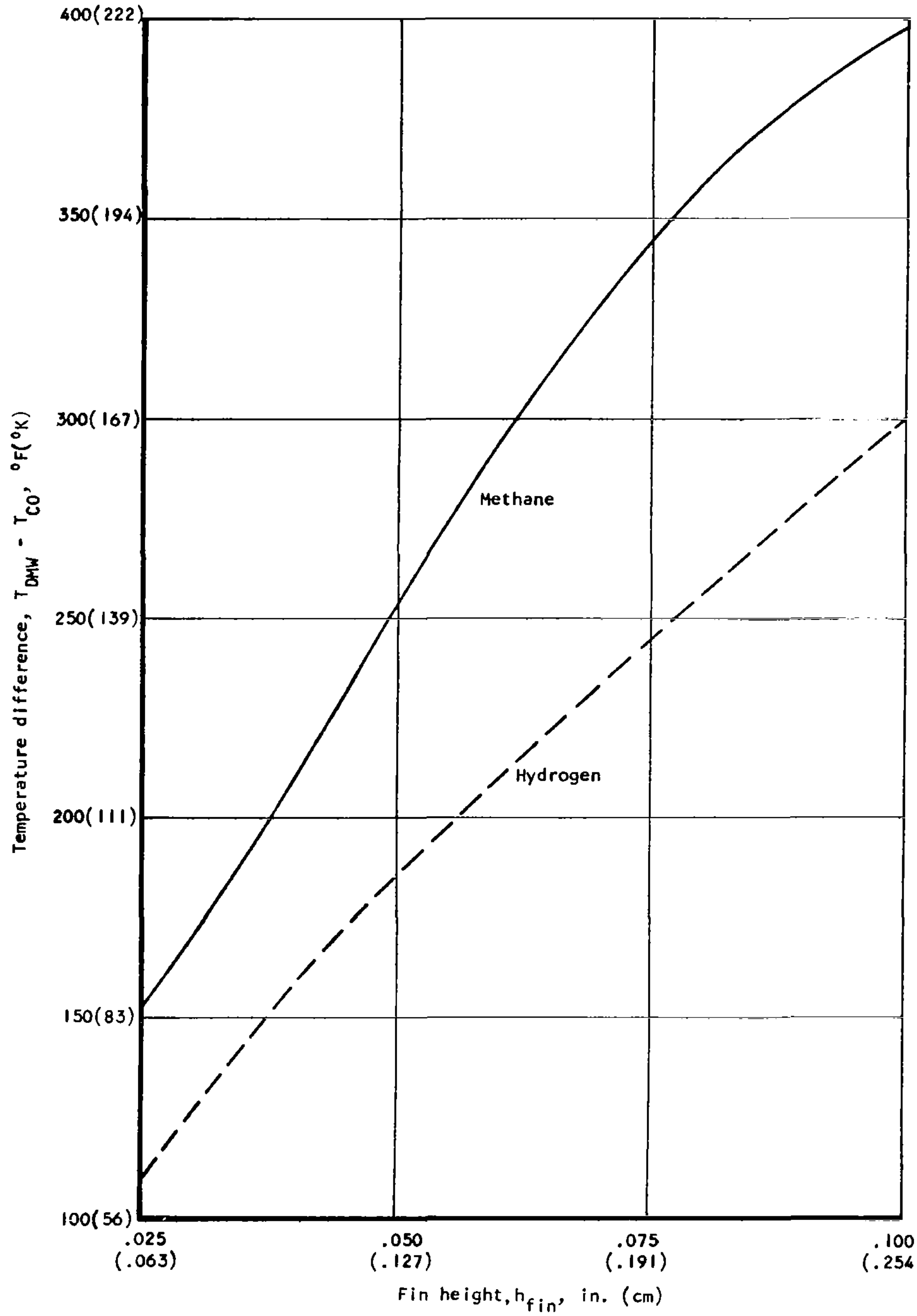

Figure 16. Effect of Fin Height on Temperature Difference at 100 Btu/sec-ft^2 (1140 kW/m^2) with T_{CO} = 1600°F (889°K) and Offset Fin Geometry, 20(7.9) RO-h_{fin} - 0.003(0.0076)

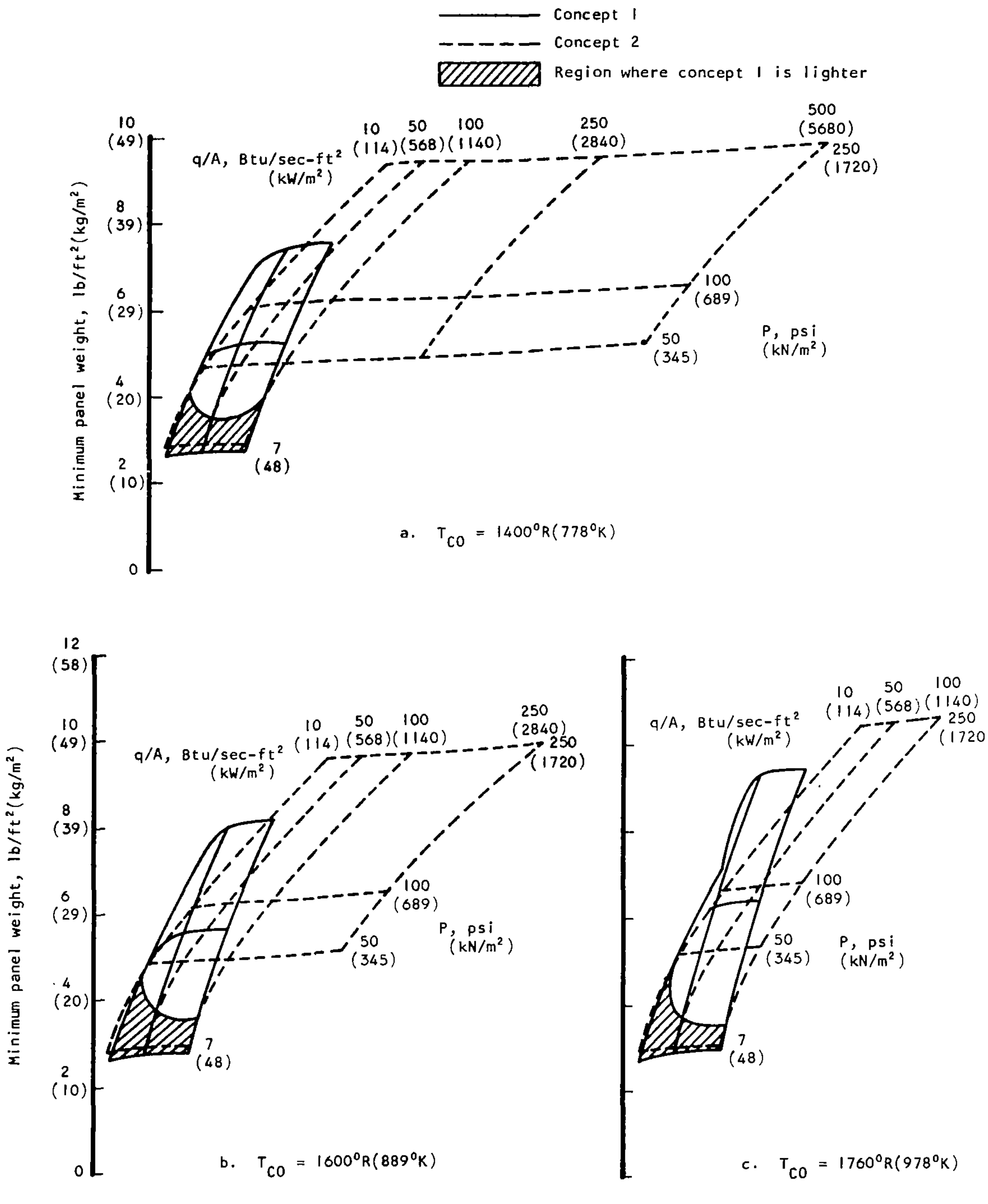

Figure 17. Comparison of Minimum Weights for Concept 1 and Concept 2 Methane-Cooled Panels at Three Outlet Temperatures with Various Heating and Loading Conditions

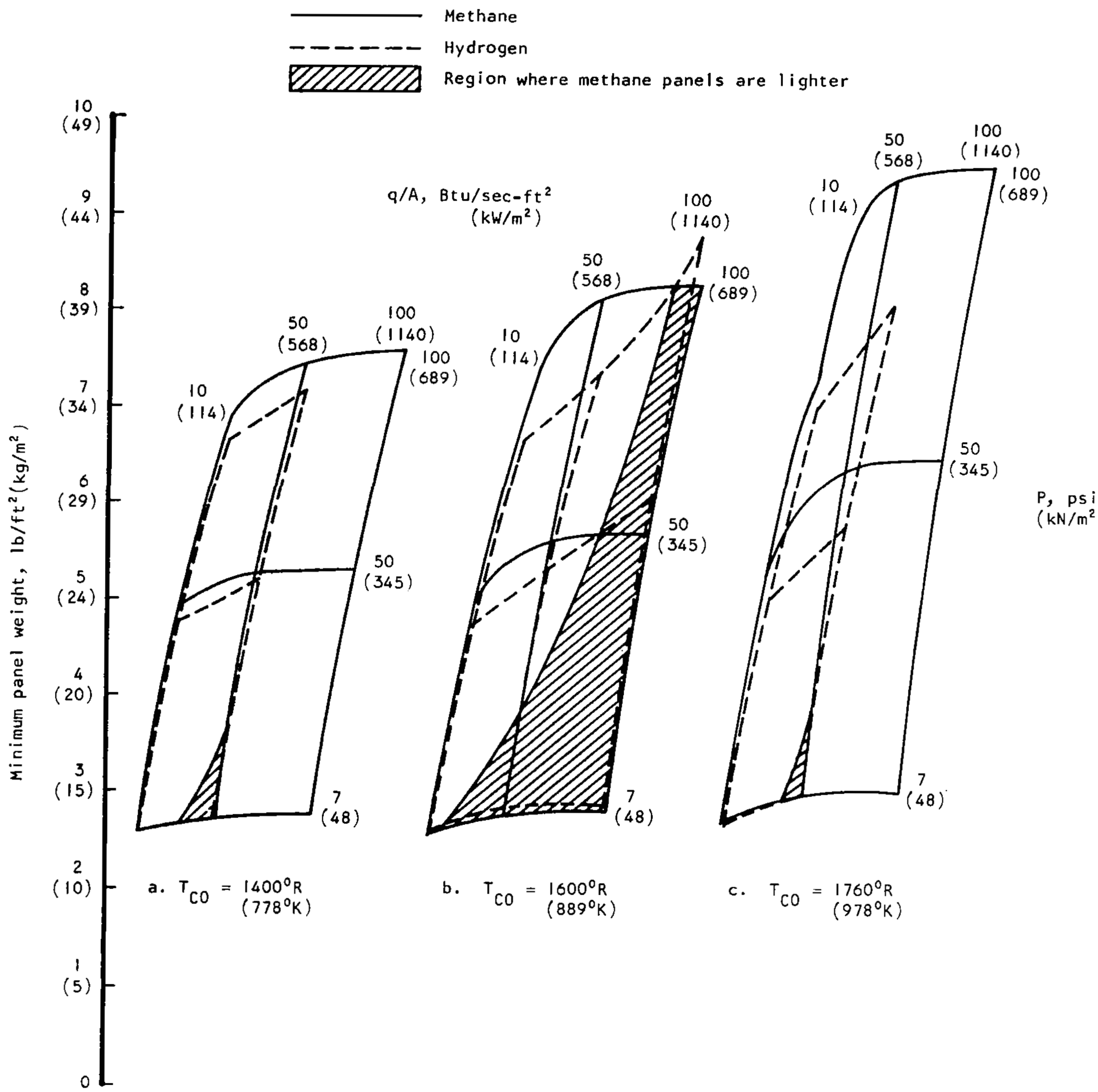

Figure 18. Comparison of the Minimum Concept I Panel Weights for Hydrogen and Methane Coolants at Three Outlet Temperatures with Various Heating and Loading Conditions

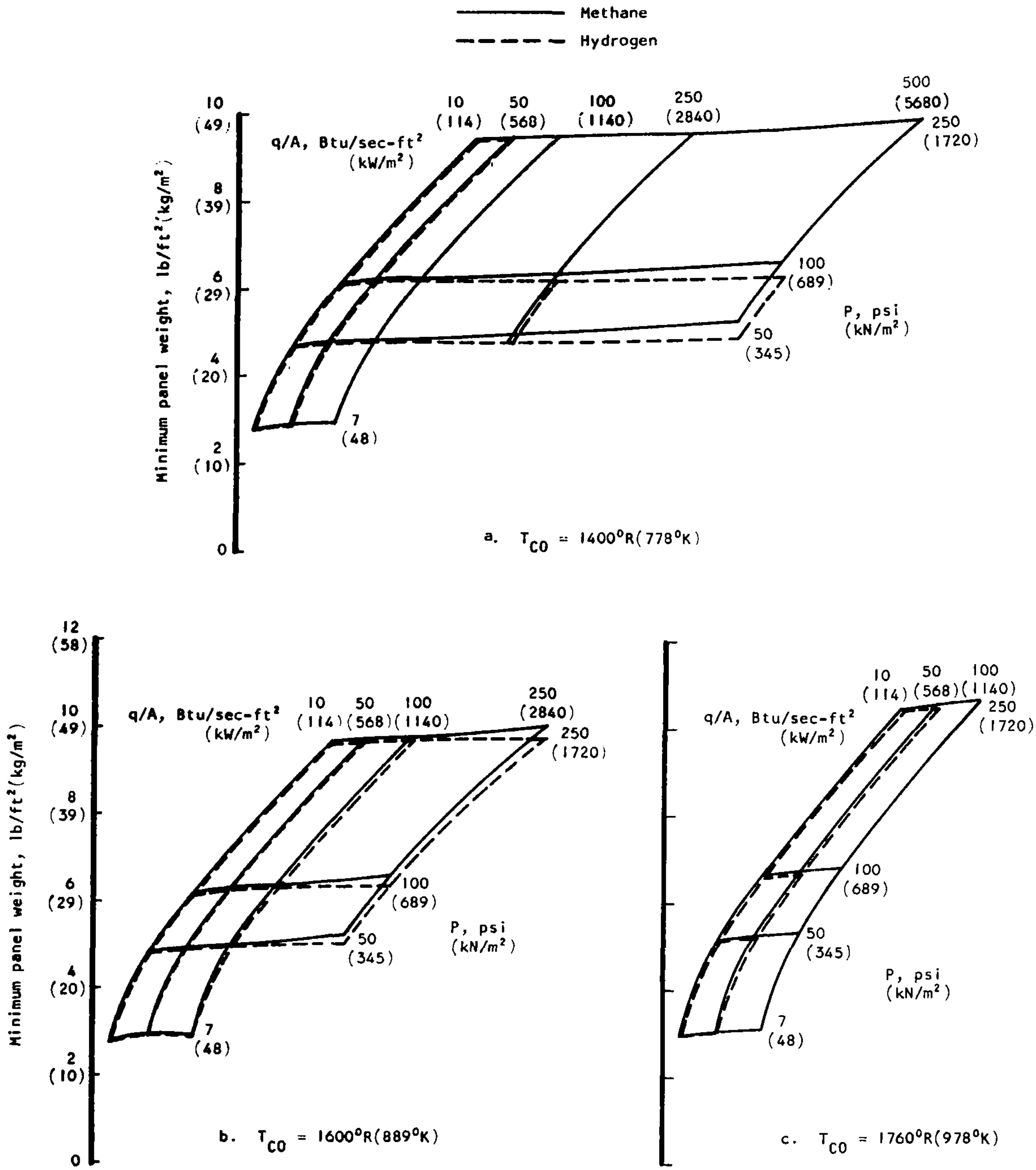

Figure 19. Comparison of the Minimum Concept 2 Panel Weight for Hydrogen and Methane Coolants at Three Outlet Temperatues with Various Heating and Loading Conditions